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DRAWING CARTRIDGE CASES*

MACHINES, DIES, TOOLS AND METHODS USED BY THE FRANKFORD ARSENAL

BY DOUGLAS T. HAMILTON†

THE difficulties met with in producing cartridge cases from sheet brass are the same as those encountered in practically all drawn sheet metal work. Admitting this, and considering the advances made in cartridge making, it would seem that a description of the methods and tools used for this work should be of general value when modified to suit the various requirements. In this article the data given refer more particularly to the drawing of cases for 0.30 caliber cartridges, the tools and methods used being taken up in detail. The practice followed by the Frankford Arsenal, Philadelphia, Pa., is interesting and presents some novel ideas in this class of work. The writer is indebted to Colonel George Montgomery for the material contained in this and subsequent articles.

Making the Cups from which Cartridge Cases are Drawn

The first operation in the making of a cartridge case is to produce a cup, then by successive redrawing operations this cup is reduced in diameter and extended to the required length. Fig. 1 shows the various steps in the sequence of redrawing operations following that of making the cup. From this illustration it can be seen that five operations are necessary to bring the case to the required length—these are called redrawing operations because the work accomplished consists in reducing and redrawing a piece that has already been drawn to cup form. The press used for making the cups from which cartridge cases are made is shown in Fig. 3. It is of the double-action type, and carries four punches and dies, thus making four cups at each stroke. This punch press operates at 100 revolutions per minute, producing 400 cups a minute, or 24,000 cups per hour. The type of blanking and cupping dies used in this machine are shown diagrammatically in Fig. 2, and also in Fig. 10 together with the tools required to make them. Fig. 2 shows the shape of the die more clearly, and gives a comprehensive idea of the action that takes place in the formation of the cup. At *A* the blanking punch is shown in contact with the top face of the brass sheet. At *B* the blanking punch has cut out a disk of the required size and carried it down to the first shoulder in the combination blanking, cupping and drawing die. At *C* the combination cupping and drawing punch has come into operation and has started to form the blank to cup shape; whereas at *D* the blank has been forced completely through the die and has been given the first drawing operation.

The sheet stock is held on the roll *A* located to the right of the machine shown in Fig. 3, and is drawn into the press under the blanking and drawing punches by means of feed rolls. After the stock passes through the rolls *B*, it is oiled by means of a rag saturated with lard oil that is contained in the tank *C*. The sheet is fed a distance equal to the diameter of the blank plus the width of the web for each stroke of the press (see Fig. 5), and after it has been started by the operator, who uses the handwheel feed illustrated, it is carried on automatically by the feed rolls located at both ends of the throat of the press. As the sheet from which the blanks have been sheared protrudes from the left-hand end of the machine, it passes through a shearing die, which in conjunction with a knife operated by a crank and connecting lever held on the extreme end of the crankshaft, cuts off the scrap, enabling it to be packed in boxes. The chief reason for cutting up the stock in this manner is to avoid having it pile up around the machine and also to enable it to be more easily removed.

The manner in which the dies and punches are laid out in order to economize in stock is shown in Fig. 5. By referring to the upper diagram in this illustration it will be seen that

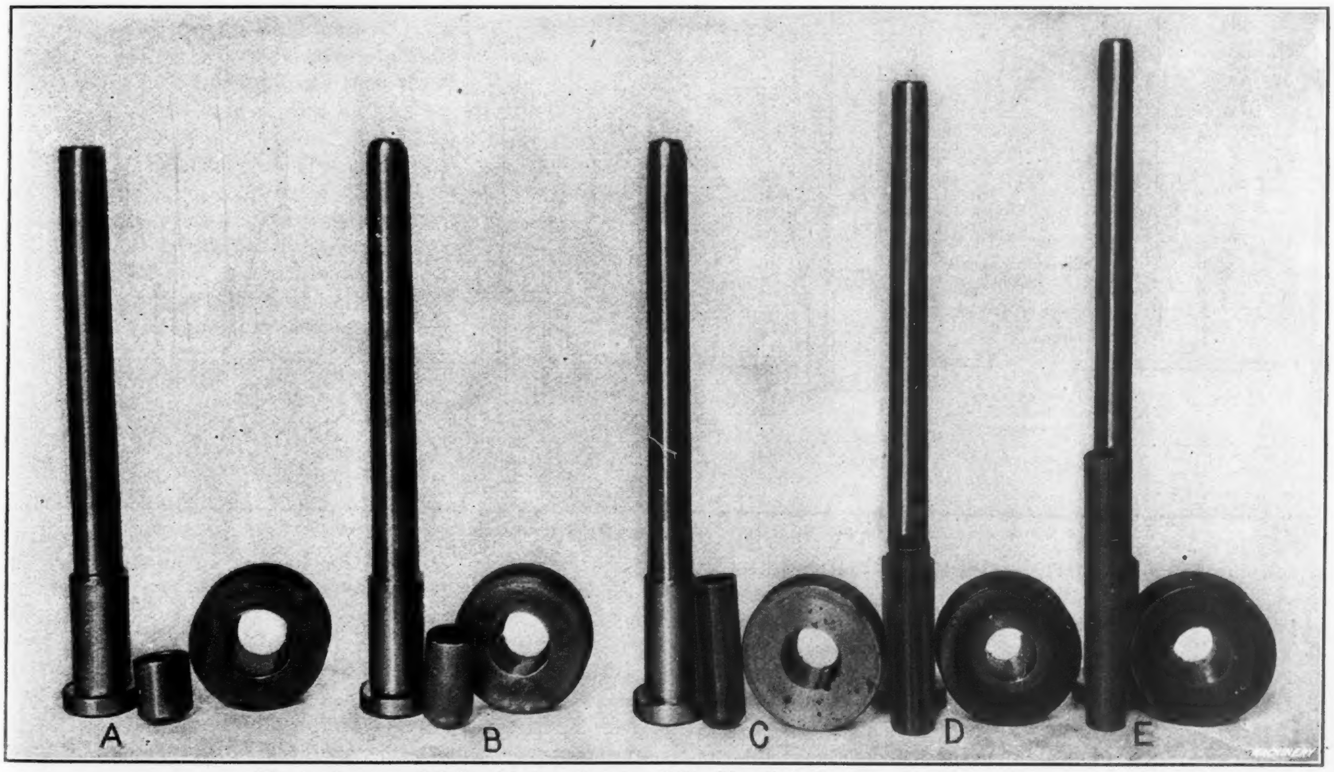

Fig. 1. Sequence of Redrawing Operations on a 0.30-caliber Cartridge Case and Tools used

* For articles on cartridge making previously published in MACHINERY, see the series on "Cartridge Making," in the March, April and May, 1911, numbers, and articles there referred to.
†Associate Editor of MACHINERY.

the centers of the four dies are located in a "diamond" shape, thus reducing the width of the sheet required and securing the most satisfactory layout for the punches and dies. The condition of the sheet after the press has made four strokes is shown in the lower portion of the illustration, which indicates the progression followed in cutting out the blanks.

Setting Drawing Punches and Dies

To set drawing punches and dies properly, requires considerable experience, as this is a difficult task under the most favorable conditions. The dies and punches are usually aligned with each other by setting them in the approximately correct position, then running through a few cups and noting the results. In the type of die-holder shown in Fig. 4, it is not necessary, of course, to reset the dies when they have been removed for grinding, if proper attention has been paid when they were first set up. This is not the case, however, with redrawing dies, as will be explained later. One peculiar point in making cups that causes considerable trouble is that it is practically impossible to produce a cup with a straight top; that is, one in which the metal is drawn to the same extent on one side as it is on the other.

The reason for a cup drawing irregularly in this manner is not due in all cases to inaccurate setting of the punches and dies, but generally to a variation in the thickness of the sheet from which the blank is cut out. It is a peculiar fact, but nevertheless true, that it is practically impossible to roll sheet metal uniform in thickness; that is to say, the sheet is thicker in the center than it is at the outer edges. The reason given for this is that the rolls, even though they be 8 or 10 inches in diameter, spring to a slight extent in the center—where they are unsupported except by their inherent strength—and thus produce a sheet of varying thickness throughout its width. Another difficulty experienced in making cups is the striking of hard spots in the metal. It is obvious, of course, that if the stock is not of a uniform hardness, the softest spot or portion will draw much more than the harder portion, and hence, a cup having one side longer than the other will be obtained. Not only will the top edge be irregular, but the walls will also be of uneven thickness. It is claimed by those who have had experience in this work that it is impossible to rectify any defect of this kind in the succeeding redrawing operations. When a cup is once started with a wall of unequal thickness, this condition prevails until the final drawing operation, so that it will easily be seen that great care must be exercised in making the walls of the cup of uniform thickness if a satisfactory product is to be obtained.

Annealing and Redrawing Operations

After the cups are made it is the general practice to anneal, wash and dry them as was described in an article on this subject in the March, 1911, number of MACHINERY. Then they are ready for the first redrawing, or second drawing operation. The temperature to which the cup is heated for annealing varies from 1200 to 1220 degrees F. The manner of handling the cups after they have been annealed, washed and dried, is to carry them in trucks, which are lifted from the floor of the annealing room to a track located above the drawing presses. These trucks are provided with false bottoms and are run along the track until they are directly over the hopper which feeds the cups to the punch press. The false bottom is then removed, allowing the cups to drop from the chute into the hopper *A* of the drawing press, Fig. 6, from which they are removed by a feeding device consisting of a wheel in which pins *B* are set at an angle of about 45 degrees with its horizontal axis. These pins are pointed, enabling the shell to be located on them, mouth first. The pins are rotated inside the hopper so that they catch the cups and deposit them in close-wound spring tubes *C*.

These tubes pass from the hopper down to the feeding slides of the drawing press (see Fig. 6) and as the shells drop out of the tubes they are caught by fingers held on the slides and carried over into line with the dies and punches. When the slides have advanced to their extreme forward positions, the punches descend and force the cups through the drawing dies, depositing them in a box located under the press. The slides are operated from the crankshaft through bevel gears and a connecting-rod that transmits power down to a horizontal shaft carrying a series of four cams. These cams contact with rollers held in the feeding slides and thus transmit the desired movement to them. The rolls are held in contact with the cams by coil springs. The machine shown in Fig. 6 operates at 100 revolutions per minute, and as four cups are drawn per stroke, it is evident that this machine has a productive capacity of 24,000 cups per hour. The drawing press shown in this illustration is used for performing the second and third redrawing operations, shown with the die and punch used at *B* and *C* in Fig. 1.

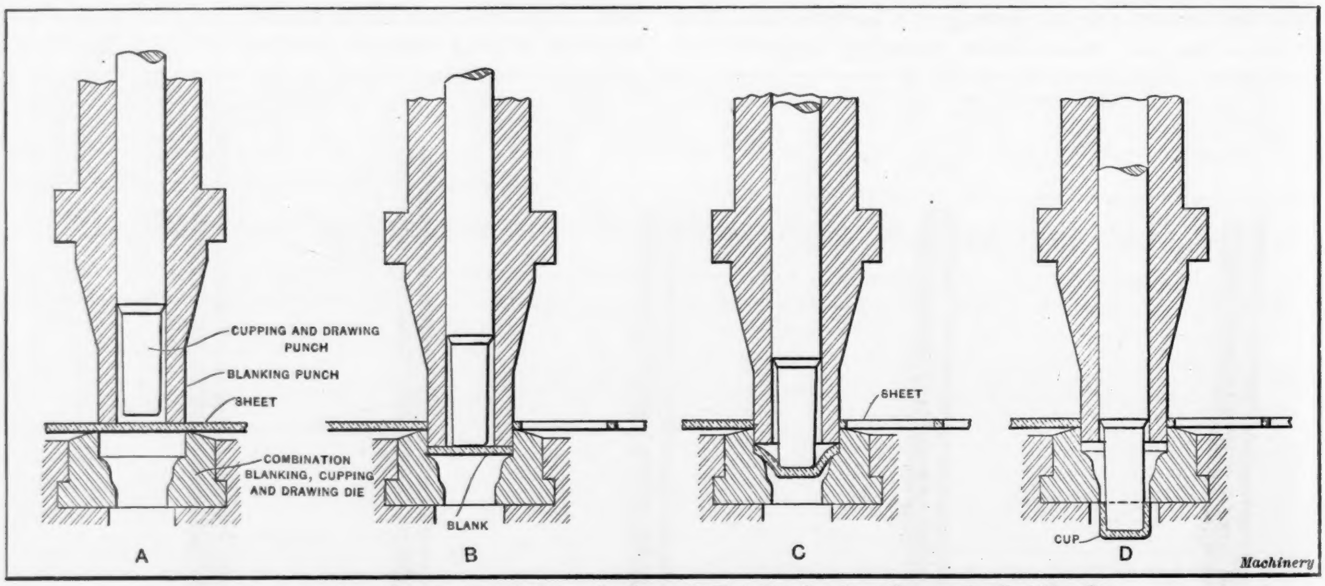

Fig. 2. Progressive Steps in the Blanking, Cupping and First Drawing Operations on the Cartridge Case

First, Fourth and Fifth Redrawing Operations

The first, fourth and fifth redrawing operations are handled in machines of a type similar to that shown in Fig. 7, which are provided with only two punches and dies instead of four, as was the case with the machine shown in Fig. 6. The feeding of the shells to the slide that carries them to the dies is practically identical with that shown in Fig. 6, but the slide is operated in a different manner. In this particular machine the slides *A* which serve as a means for carrying the cups from the feeding tubes *B* over into line with the drawing dies are actuated in their movement by means of a bellcrank lever receiving power from a cam held on the crankshaft *C* of the press.

While the shells are fed to the punch with the mouth up, it sometimes happens that one will pass down the feeding tubes to the slide the wrong way, that is with the bottom up. Now should such a shell be allowed to pass over into

line with the die, it would mean that the punch would be broken and the die either broken or damaged to such an extent that it would be unfitted for use. It is not uncommon also to have shells pass down to the slide that are dented or otherwise defective which would prevent them from feeding into the die properly. Should such a shell pass down the feeding tubes and stick in one of the slides, it would mean that the punch would come down on the slide and break, not only putting the machine out of commission for a time, but perhaps causing serious damage to the attendant as well.

In order to provide against such accidents, Mr. August A. Plate, general foreman, invented an ingenious tripping device

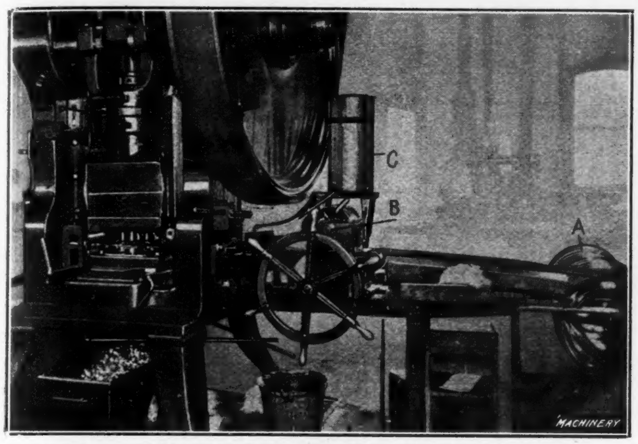

Fig. 3. Making the Cups, Four at one Stroke of the Press, at the Rate of 24,000 per Hour

that is applied to this machine and works very satisfactorily. This device, while comparatively simple in construction, is positive in its action, and has been the means of saving a lot of money in the cost of dies and tools. It also enables one attendant to run four instead of two machines. Essentially, this device consists of a projecting stud held in the crankshaft of the press, and which when the feeding slide is operating normally passes through a slot cut in a lever that is connected with the bellcrank lever operating the slide.

Now, if for any reason the slide should be prevented from making a complete forward or backward stroke, this projecting pin would not pass through the slot in the lever men-

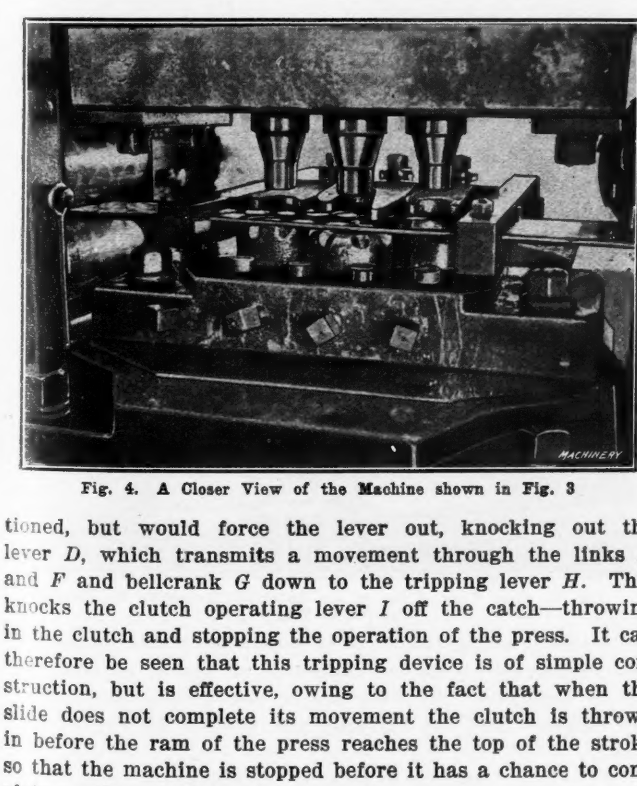

Fig. 4. A Closer View of the Machine shown in Fig. 3

tioned, but would force the lever out, knocking out the lever *D*, which transmits a movement through the links *E* and *F* and bellcrank *G* down to the tripping lever *H*. This knocks the clutch operating lever *I* off the catch—throwing in the clutch and stopping the operation of the press. It can therefore be seen that this tripping device is of simple construction, but is effective, owing to the fact that when the slide does not complete its movement the clutch is thrown in before the ram of the press reaches the top of the stroke so that the machine is stopped before it has a chance to complete another stroke.

Final Redrawing Operation

The fifth redrawing operation is accomplished in a press similarly equipped to that shown in Fig. 7. These presses operate at 100 revolutions per minute, and turn out 12,000 cups per hour. Several annealing operations take place between the time when the cup leaves the first redrawing operation and the time when it is ready for trimming, but as these have been described in the article mentioned no reference need be made to them here. Before the fourth redrawing operation is accomplished the shells are taken to a heading machine of the horizontal type where they are "bumped." This operation is accomplished in order that in the successive redrawing operations the head of the shell will not be reduced too much in thickness.

The 0.30 caliber cartridge case has what is known as a solid head; that is, the top portion of the shell that contains the primer is not indented to form a pocket for the primer, the pocket itself being simply a hole forced into the head. This type of cartridge has been found necessary for use with smokeless powders. The former method used in making 0.30 caliber cartridge cases was to form the pocket by forcing in the head which was very little thicker than the sides of the shell near the head. This construction, however, was found to be too weak for smokeless powders, as the head would blow off. The "bumping" is a very simple operation and is somewhat similar to heading except that the punch is perfectly flat and simply gives the shell a blow, upsetting it slightly and flattening it so that in the two final

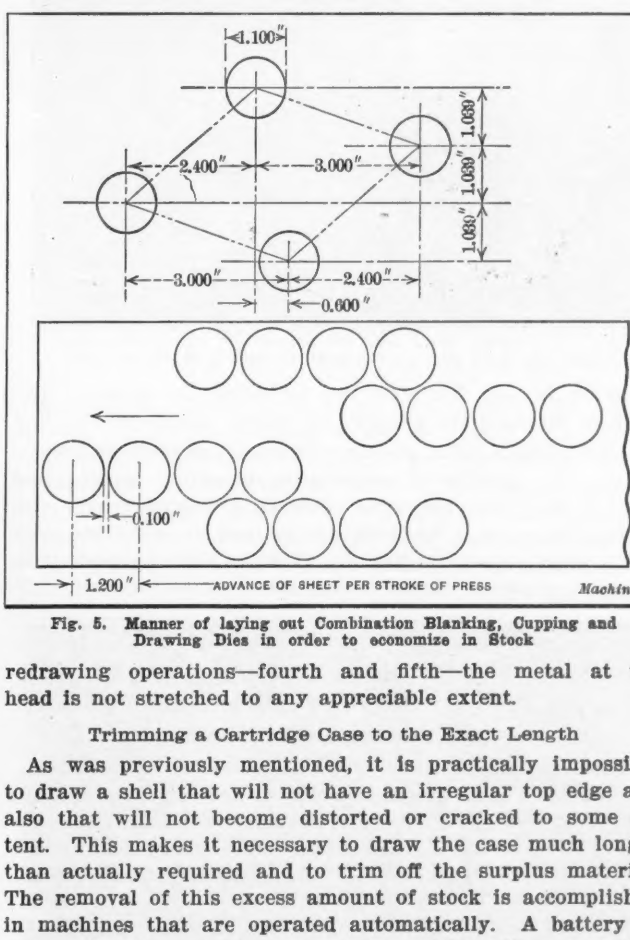

Fig. 5. Manner of laying out Combination Blanking, Cupping and Drawing Dies in order to economize in Stock

redrawing operations—fourth and fifth—the metal at the head is not stretched to any appreciable extent.

Trimming a Cartridge Case to the Exact Length

As was previously mentioned, it is practically impossible to draw a shell that will not have an irregular top edge and also that will not become distorted or cracked to some extent. This makes it necessary to draw the case much longer than actually required and to trim off the surplus material. The removal of this excess amount of stock is accomplished in machines that are operated automatically. A battery of these trimming machines at work on 0.30 caliber cartridge cases is shown in Fig. 8, while Fig. 9 shows a closer view of one of the machines and gives a clear idea of its working mechanism. As will be seen upon reference to Fig. 8, these trimming machines are arranged in such a manner that the various hoppers can be filled from an overhead conveying system. This arrangement consists of a track similar to that used in the drawing press department previously referred to, and enables one man to attend to an entire line of presses. The track accommodates a truck in which the shells are carted along the line and from which they are ejected through a false bottom, dropping into the hoppers located over the machines. The feeding of the shells down to the trimmer is accomplished by the same type of hopper as previously described, but the subsequent handling is somewhat different. As the shell descends from the hopper it passes through a

locating cage *A* from which it is carried forward by a plunger *B* and is located on the cutting-off punch *C*. Here it is held by friction while a circular trimming tool *D* advances and trims off the surplus stock. The shell and trimming are then ejected from the cutting-off punch by a sleeve *E* operated from the left-hand end of the machine, the shell being deposited in one box and the trimming in another; two separate channels are provided as shown clearly in Fig. 9.

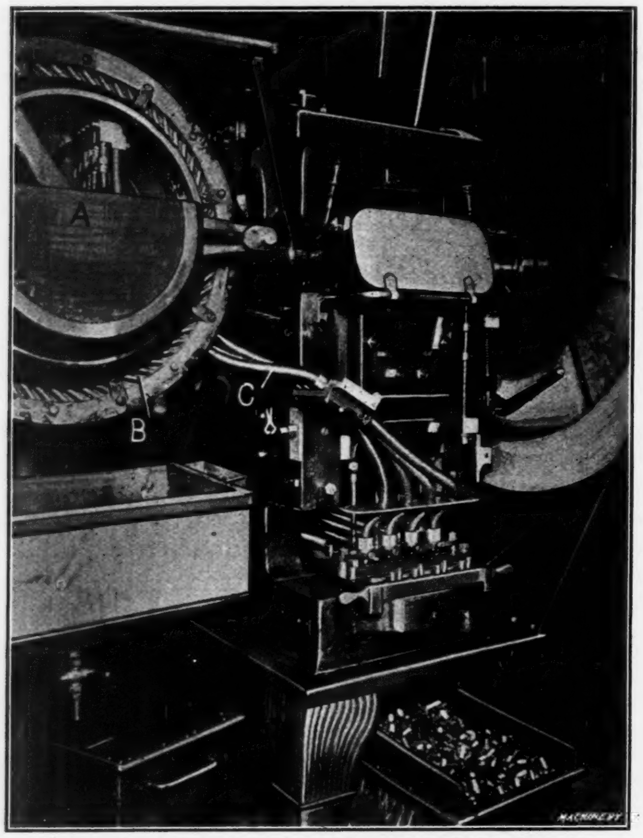

Fig. 6. Drawing Press used for Second and Third Redrawing Operations—Four Cups per Stroke at the Rate of 24,000 per Hour

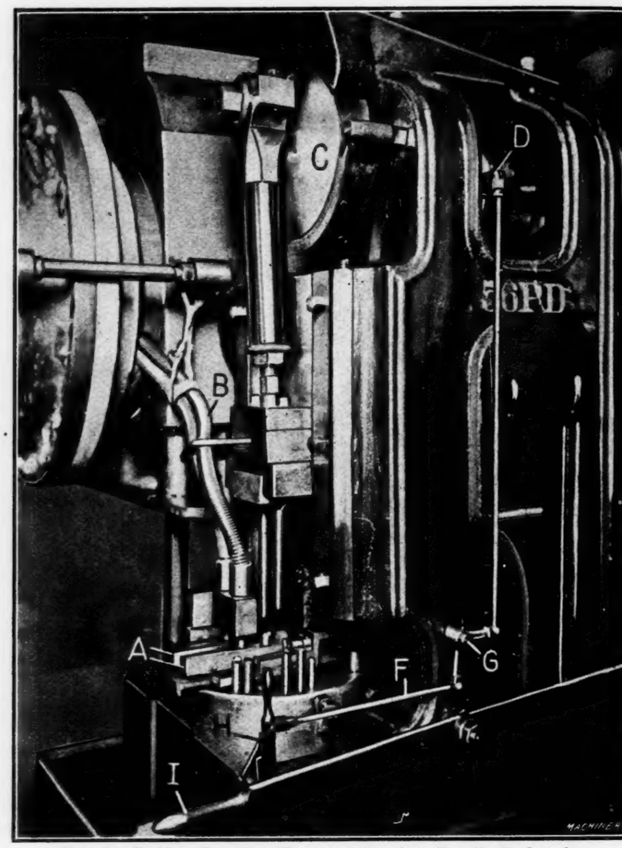

Fig. 7. Duplex Drawing Press performing Fourth Redrawing Operation and turning out 10,800 Cups per Hour

Making Combination Blanking, Cupping and Drawing Dies

While the making of combination blanking, cupping and drawing dies does not differ materially from ordinary tool making, there are a few points in connection with this work that it might be well to explain. The die blank is made from a special grade of Firth-Sterling steel containing from 1.11 to 1.30 per cent carbon. This is an extremely high carbon steel, but has been found satisfactory for this class of die owing to the great wear that it is subjected to when in use. The first step, of course, in making one of these dies is to cut off the blank from a bar of stock, and then by means of drills, reamers, etc., to shape the hole in the die to the correct form. Fig. 10 shows one of these dies at *C* together with a blank and a cup made from it, while *D* is the combination cupping and drawing punch and *E* the blanking punch.

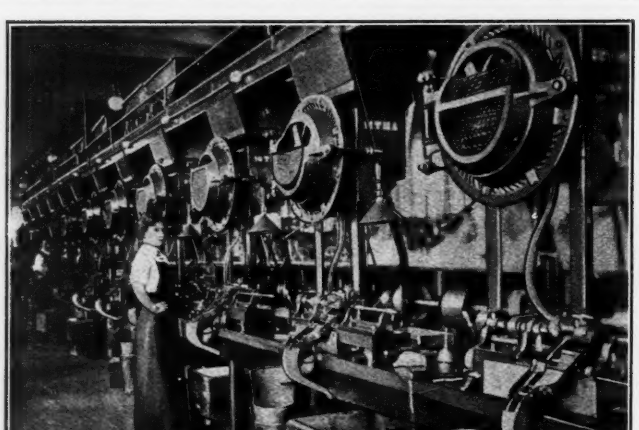

Fig. 8. A Battery of Automatic Trimming Machines at work on 0.30-caliber Cartridge Cases

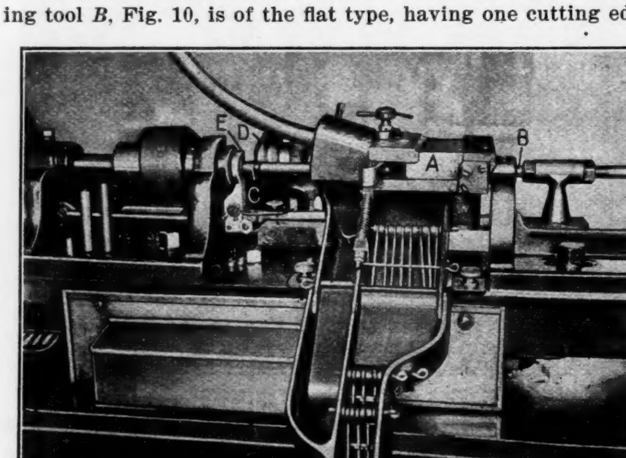

Fig. 9. Close View of One of the Automatic Trimming Machines shown in Fig. 8

Fig. 11 shows a toolmaker completing one of these dies, and Fig. 12 shows a close view of the chuck used for holding it. The manner in which the die is held while being drilled, counterbored and reamed is identical with that used when it is set up in the press, so that in this way the conditions in both cases are as nearly alike as possible. The chuck consists of a female center *A* in which a recess is provided that fits the external body of die *B*. The outer end of this center is turned and threaded to fit a cap *C* that is machined to correspond with the other end or smallest diameter of the die *B*. This cap holds the die rigidly in position while it is centered with the female member *A*. In addition to a drill, and, of course, a boring tool to true up the hole, two tools are used for finishing the hole in this die. The first or roughing tool *B*, Fig. 10, is of the flat type, having one cutting edge, while the finishing tool *A* is of somewhat similar shape but has considerably more circumference so that a rounder hole will be produced. The first tool is used merely for roughing out purposes and for bringing the hole to its approximate shape.

Making Redrawing Die Blanks

The blanks for redrawing dies are turned out in Cleveland automatic screw machines and are ready for the final reaming operation when they drop from the machine. This method is commendable in that it reduces the cost of the dies

to a minimum. The steel used for making these dies is a special grade of Firth-Sterling steel containing from 1.11 to 1.30 per cent carbon, and for the particular die *G* illustrated in Fig. 13 a bar 1¾ inch in diameter is used. This die is a second redrawing operation, die for a 0.30 caliber cartridge case, and is 1 23/32 inch in diameter by 1/2 inch thick. In order to have the hole true with the external diameter, great

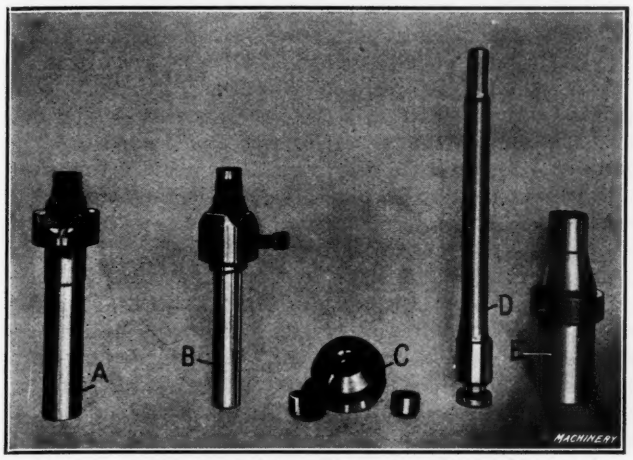

Fig. 10. Combination Blanking, Cupping and Drawing Punch and Die, and the Tools used for making the Die

care is taken in spotting the work and then removing the hole entirely from the next blank, using a fairly wide cut-off tool. The order of operations accomplished in the proper sequence is as follows. First, feed stock to stop *A*; second, turn external diameter with box-tool *B* held in turret and spot with a drill *C* retained in the same holder; third, drill with drill *D*; fourth, ream straight portion of hole with

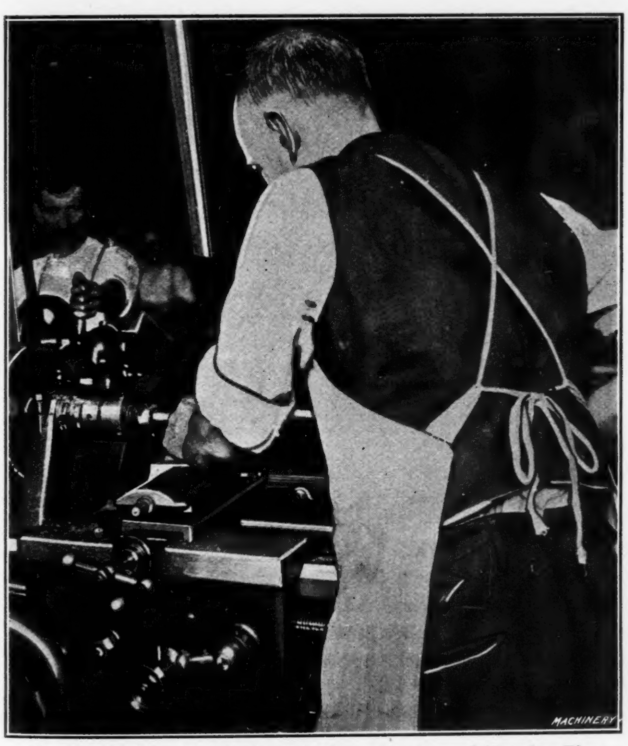

Fig. 11. Making Combination Blanking, Cupping and Drawing Dies

reamer *E*; fifth, bell-mouth with reamer *F*, and face with a tool held on the rear cross-slide; sixth, cut-off with a tool held on the front cross-slide.

The drawing dies, after being rough-formed in the manner illustrated, are then taken to the tool-room where they are reamed out to the exact diameter and bell-mouthed to the correct shape, after which they are ready for hardening. For hardening, the dies are heated in a muffle furnace to a temperature varying from 1400 to 1450 degrees F., and are then "spouted" as illustrated in Fig. 14. The spouting of the die consists in directing a stream of water through the hole in order to harden it and at the same time leave the external diameter practically soft. The reason for this is that the die, after hardening, is not drawn, and if the entire blank were hardened it would break very easily. Having the external diameter soft increases the strength to a considerable extent, and the dies wear much longer and do not break as easily.

When spouting, the die is held in a cage formed in the base of the bracket *A* and then the funnel *B* (which is similar in shape to an ordinary oil funnel except that the lower tapered tube is left off) is placed over it, the water being directed through this funnel and thence to the hole in the die. The funnel is provided with a handle to enable the operator to place it quickly in position over the die after the latter has reached the proper temperature, and has been placed in the fixture. The operator removes the die with a pair of tongs, holding the tongs in one hand and the funnel in the other.

Great care must be taken in heating this grade of steel because of its high carbon content. A variation of 10 degrees F. one way or the other from the temperature found most satisfactory will in many cases make the die defective for the operation that it is to accomplish. It has also been found necessary to heat the dies for various operations to different

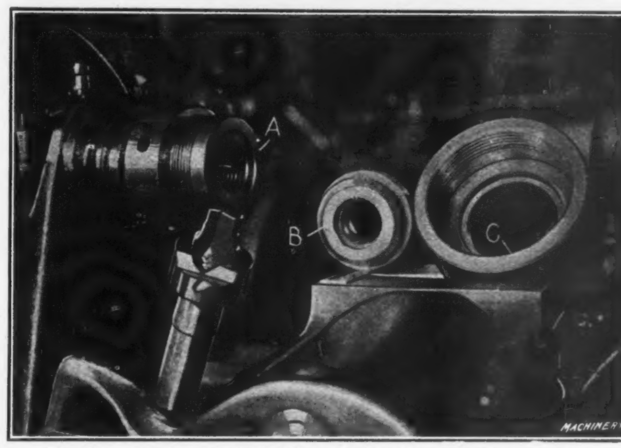

Fig. 12. Chuck used for holding Combination Blanking, Cupping and Drawing Die Blank when machining the Hole

temperatures; that is to say, the die that would be used for a first redrawing operation would be heated to a different temperature from one that would be used for the fifth redrawing or final drawing operation. The reason for this is that the pressure exerted on the die, by forcing the cup through it with the punch, on the first drawing operation, is much greater than that for the final drawing operation, and hence the die cannot be as hard and must have a more porous grain to withstand the additional pressure.

Lapping Redrawing Dies

After hardening, redrawing dies are lapped in the manner illustrated in Fig. 15. The die is held in a holder, resembl-

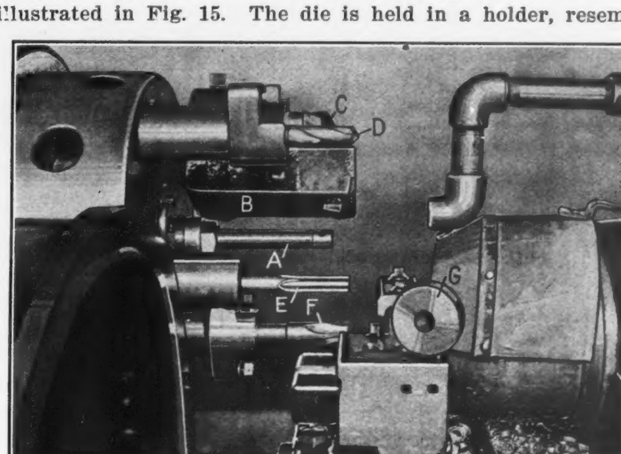

Fig. 13. Making Redrawing Die Blanks in a Cleveland Automatic Screw Machine

ing somewhat in shape that used in the drawing press, which is held in the chuck of the speed lathe. For lapping, a lead lap is used. This is held on a steel plug provided with a handle driven through it at right angles to the lapping portion, so that the operator can grip it with both hands and thus hold the plug in the proper position in relation to the

hole in the die. The speed lathe in which the die is held is operated at from 2000 to 2400 revolutions per minute, and the spindle is always kept a snug fit—also there must be no end play. A mixture of lard oil and No. 10 emery is used for lapping, this being found best for both the roughing and finishing operations. The mouth or bell-mouthed portion of the die is lapped with emery cloth of the same grade—No. 10. The lapping of a drawing die is an operation that must be very carefully handled. Not only must the hole be of the correct size, but the radii of the bell-mouthed portion must be exact. The correct lapping of the die is more a matter of experience than anything else, and it is practically impossible to give any definite information on the subject. One point, however, that should never be ignored is the fact that the lap should always be presented in a line parallel to the axis of the die. If it is tilted over the least bit to one side or the other a hole will be produced that will not only be out of round, but that will not be straight; that is, if the die were placed on the arbor it would be found to run untrue because the hole would not be exactly in the center of the blank on both sides.

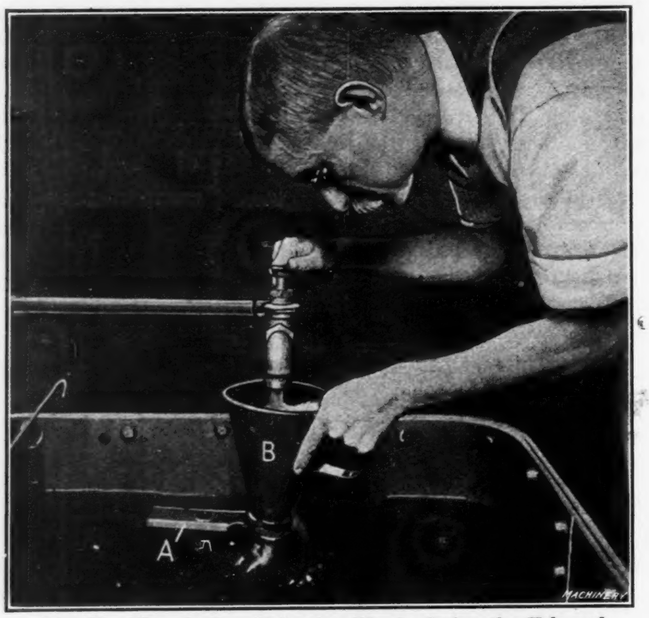

Fig. 14. "Spouting" a Redrawing Die—hardening the Hole and leaving the Exterior comparatively Soft

After the dies are lapped to the correct size and shape, they are ready for use in the press and are then turned over to the drawing press department. Drawing dies for all redrawing operations up to the final operation are used until they have worn approximately 0.0017 inch large. When they have become worn to a size this much greater than the actual diameter of the cup required, they are taken out of the press and annealed. The dies are then reamed out to the next size larger—that is for the previous redrawing operation than that for which they were originally made—and used over again. This is repeated until the dies have been used five times. Redrawing dies made from Firth-Sterling steel of the carbon content previously given are good for making 40,000 cups before they have become worn too large. The peculiar point about this steel is that it does not warp out of shape in hardening and also does not produce any scratches on the work. It is of extremely fine grain, hardens well and produces a shell free from scratches and other imperfections. The only thing that makes it unfit for use is when it becomes worn too large. Otherwise the condition of the hole in the die is as good at the completion of 40,000 cups as it was when first used.

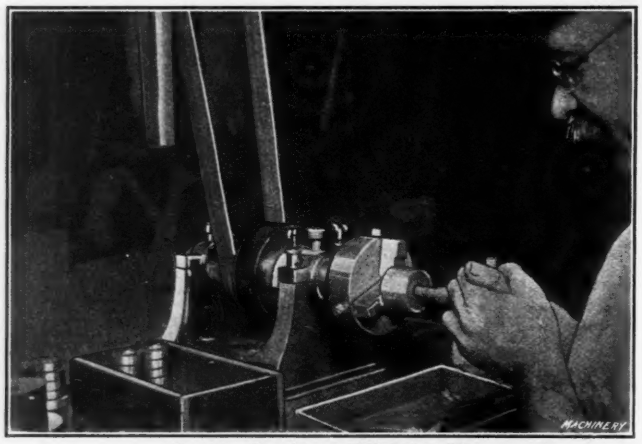

Fig. 15. Lapping a Redrawing Die

Making Redrawing Punches

Redrawing punches are also made from Firth-Sterling steel, but the carbon content for the punch is considerably lower than that used for the die, and never should exceed over 0.60 per cent. There are a few points that require careful consideration in making drawing punches. In the first place, the stock should be centered as true as it is possible to get it. There is a good reason why this operation should be carefully done. If the piece of stock from which the punch is to be made has not been centered true, the finished drawing punch when hardening will be bent out of shape. The reason for this is that when a bar that has been incorrectly centered, so that it runs eccentric, is turned, more stock is removed from one side than the other, and the turning, instead of being parallel with the grain, cuts across it; hence the ability of the steel to resist deflection in hardening is not as great as it would be had the support not been removed from one side. The explanation given for this is that in rolling bar stock the fiber or grain of the metal is drawn out in practically a straight line and when this condition does not exist in the finished article distortion takes place, because, in cooling, the fibers of the stock revert to their original positions parallel with the axis of the bar.

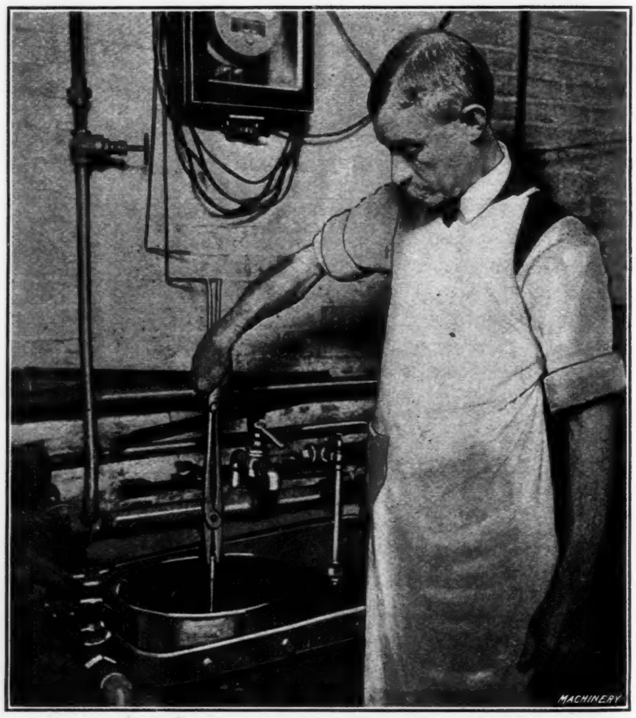

Fig. 16. Hardening a Redrawing Punch—Note Method of Dipping

Another point that is of considerable importance is that never less than 1/32 inch of material should be removed from the bars if the finished piece made from it is to be hardened. There is a certain decarbonizing portion surrounding a bar of stock that prevents the steel from hardening properly, and this decarbonizing portion should always be removed from those parts of the punch that must be hardened; if not, soft spots will be experienced.

The drawing punch should be heated in a muffle furnace very slowly until it has obtained the correct temperature, and while being heated it should be constantly rotated to prevent warping. The temperature to which drawing punches are heated varies from 1400 to 1425 degrees F. They are quenched in a bath consisting of 15 parts water and 1 part common potash, and are dipped in a vertical position as illustrated in Fig. 16.

METHODS OF CHUCKING BEVEL GEARS

ACCURATE AND EFFICIENT METHODS DEVELOPED BY THE HEALD MACHINE CO., WORCESTER, MASS.

The Heald Machine Co., Worcester, Mass., has given the problem of chucking both spur gears and bevel gears, preparatory to grinding, a great deal of study during the past few years, and has devised, at different times, what have been considered very accurate and efficient methods of chucking. There is a great diversity of opinion among the users of internal grinding machines in regard to what seems to be the most desirable method of holding or chucking gears for grinding; furthermore, a device that is very successful in one shop does not always appeal to the man in charge of another shop doing similar work. It seems to be very difficult, in many cases, to get the men responsible for results in different shops, to appreciate the advantages of a method which is different from what they have used in the past, or, in some instances, even to consider carefully the exact advantages of one scheme as compared with another, for securing the accurate results required in grinding operations.

Some of the successful methods of chucking gears which have been developed by the Heald Machine Co. are shown in the accompanying illustrations. Fig. 1 shows an excellent method of holding crown gears or ring gears such as are used in automobile transmissions. An unhardened master gear *A* is mounted on the faceplate so as to run true, and certain teeth are cut away at the inner ends as shown at *B*, leaving three sets of three or four teeth each as at *C*, which are longer than the others. When a gear to be ground is set into this master gear it will be centered with great accuracy and run true. The gear can be held in position very easily by two or three spring clips *D*. There is only one position that the gear to be ground can take with this method of holding, and as the master gear has contact points practically 120 degrees apart, the gear will not rock on its support, and extremely accurate work may be produced.

Fig. 2 shows a standard draw-in collet fitted with special jaws for holding bevel gears, these jaws being arranged to bear at the bottoms of the tooth spaces. Jaw *A*, as will be seen, has a slight projection *B* which extends over the outside of the gear to prevent it from sliding out as the jaws are closed; hence, by simply opening and closing the jaws of this collet the gear is clamped and released and no straps or yokes are necessary to hold the work in position. An objection to this design is that if the chuck teeth do not have a good bearing surface at the bottom of the tooth spaces, a chance of error will be introduced.

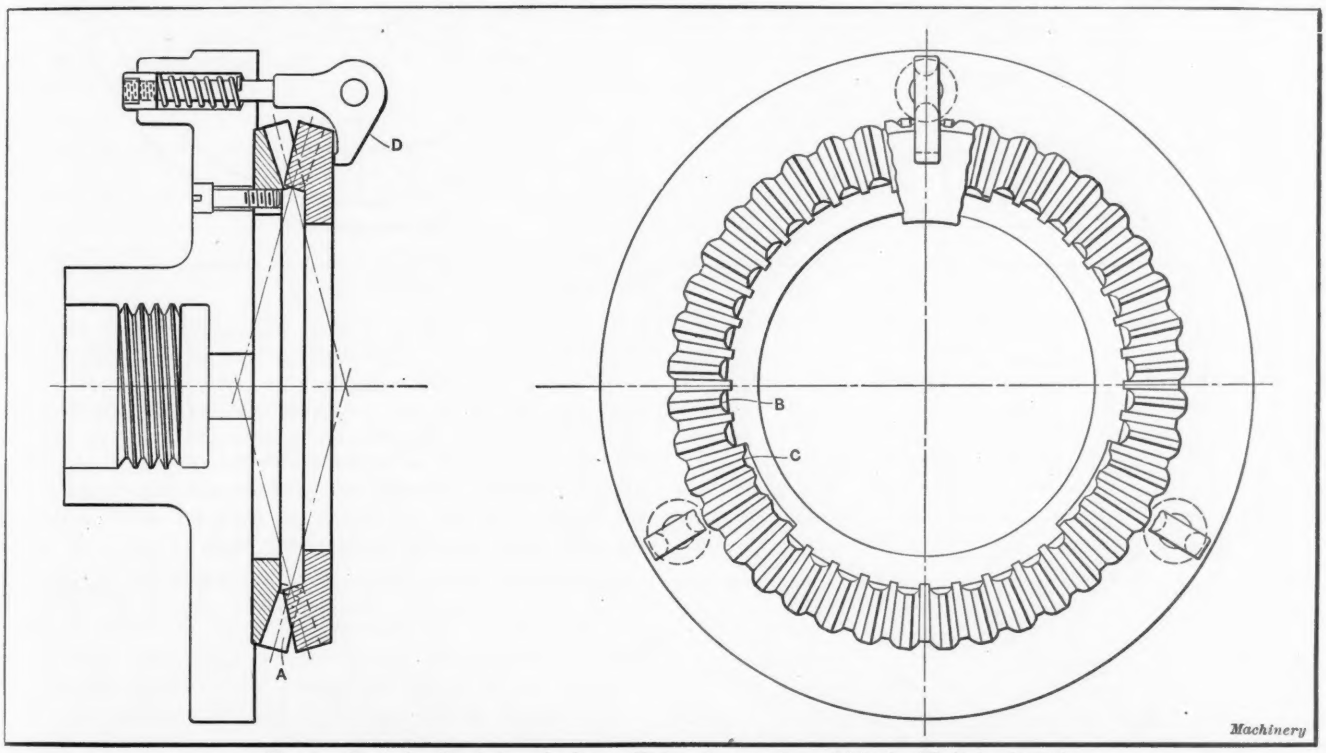

Fig. 1. Chuck for Bevel Gear having a Master Gear against which the Work is clamped

Fig. 3 shows a method of holding bevel gears by the use of taper rolls which enter between the teeth, giving a pitch-line control method of chucking. This illustration shows a chuck having six rolls arranged in three pairs. These rolls, for gears of other sizes, may have one or possibly two teeth between them, according to the number of teeth in the gear to be chucked. This gives practically a six-point contact, and yet the rolls are so disposed that the gear will be held solidly in the chuck with less tendency to rock than if they were evenly spaced around the circumference. The use of six rolls in three pairs, instead of three rolls evenly divided around the gear, gives superior results because the double rolls furnish a sort of check on each other in locating the gear. They also provide additional wearing surface, thus increasing their durability, and, according to the experience of the Heald Machine Co., gears that run very accurately are produced when this form of chuck is used.

The chuck has a faceplate *A* which screws onto the nose of the grinder spindle *B*. In front of the faceplate *A* there is a second plate *C* recessed to a suitable angle for receiving the gear *D* and rolls *E*. The rolls are held by triangular shaped plates *F* and cap-screws. These plates and the cap-screws hold the rolls back firmly against the chuck plate *C*, which prevents dirt from getting under the rolls, but allows a slight freedom for the rolls so that they can adjust themselves slightly with relation to the gear teeth. The gear is held in place by a yoke *G* and thumb-screw as shown. This yoke exerts a pressure in a line parallel with the axis of the gear, and assists in forcing the gear back to a solid and concentric location.

It will be noted that the surface on part *C* is marked "Finish for indicator," and it has been suggested that after the angular surface is accurately ground true as to size and position, this outer surface also be ground true; then if it should be necessary to remove the fixture from the machine or exchange it for another size, when the fixture was replaced it would be a simple matter to test its accuracy by applying an indicator to the finished surface previously referred to. If any error were observed, the clamping screws which hold parts *A* and *C* together would allow for whatever adjustment might be necessary to enable part *C* to run true, without going to the trouble of re-grinding the surface for locating the rolls. While it would cost a little more to make this division between parts *A* and *C*, this cost would be

saved many times over because plate *A* could be arranged to receive many different types of chucking devices, if the user of the grinding machine had a variety of gears differing as regards the number of teeth and pitch.

According to the experience of the Heald Machine Co., the rolls used for chucking bevel gears should be tapering because they make contact with curved surfaces, the elements of which converge at a common vanishing point. Some mechanical men who stand high in the profession of making gears claim that cylindrical rolls are just as satisfactory as tapering rolls, although this company considers the tapering roll superior, and is inclined to believe that those who consider cylindrical rolls satisfactory, are using them in connection with gears having a face width that is small in proportion to the gear diameter, as, for example, ring gears used in automobile transmission. In such cases the error does not seem to be as pronounced, but with miter gears, especially when the face width is say one-quarter to one-fifth the gear diameter, there appears to be a most decided tendency for the gear to rock on a cylindrical roll at a point about midway of the face of the teeth.

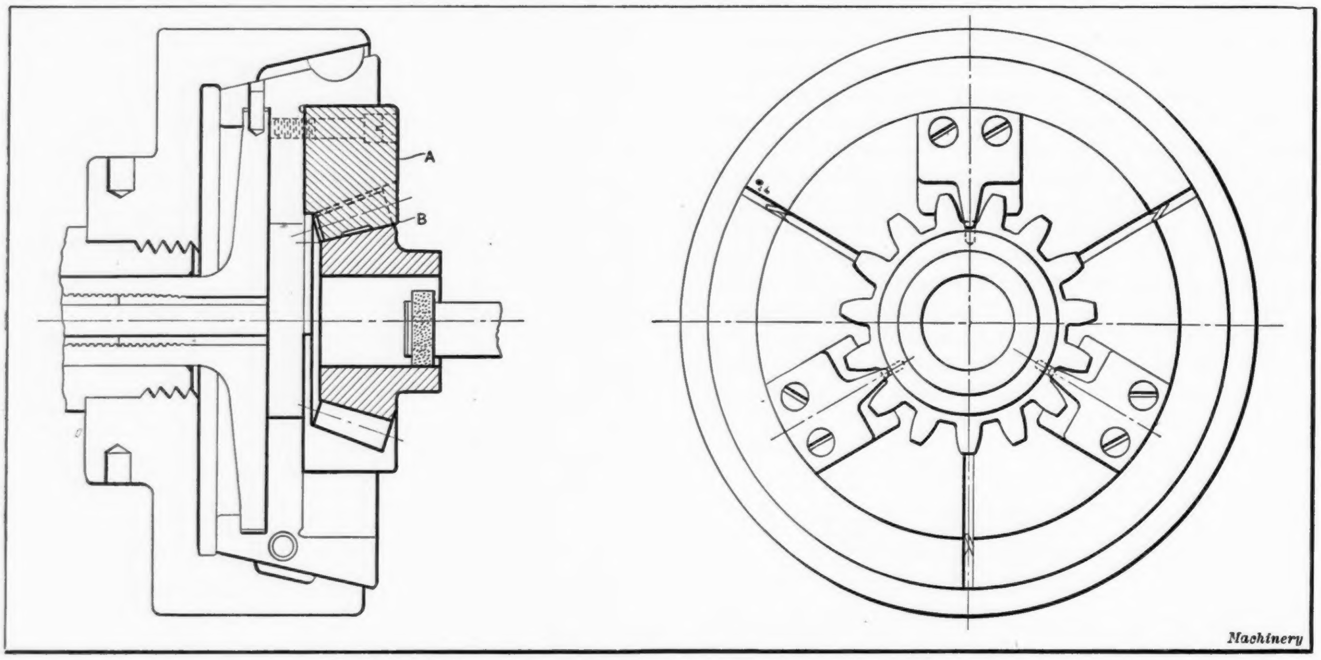

Fig. 2. Bevel Gear Chuck with Jaws which engage Bottoms of Tooth Spaces

The method of calculating the diameter and taper of the rolls is to select for the large end of the roll a diameter which will bring the surface of the roll about 1/16 inch above the outside diameter of the gear teeth. The taper of the roll should be approximately that which would cause the roll, if extended, to converge at the apex of the pitch cone. If these rolls are to make proper contact with the tooth surfaces, the points of contact must lie in a line running from the large end of the tooth to the vanishing point of the pitch cone, and if two elements in the surface of the roll meet in a common vanishing point, all other similar elements must apparently meet at the same point, indicating that these rolls should be part of a true cone. Experience seems to show this to be true, although the center of the roll is slightly outside the pitch circle of the gear. Working on this basis, it is not difficult to lay out, with considerable accuracy, the included angle of the rolls and obtain the angle of the surface in the chuck plate against which the rolls bear. In actual manufacturing, this angle can be checked easily by simply inserting the taper rolls between the teeth on opposite sides of the gear, thus forming a temporary gage.

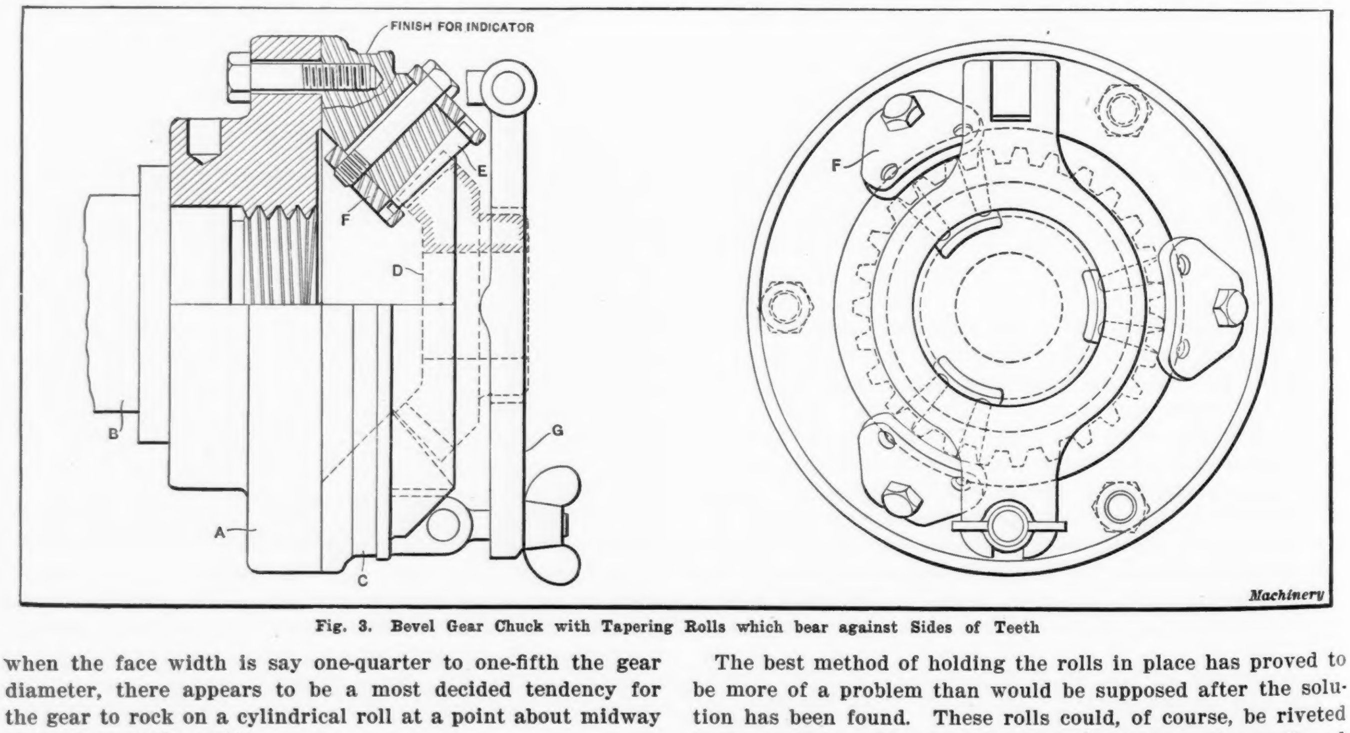

Fig. 3. Bevel Gear Chuck with Tapering Rolls which bear against Sides of Teeth

The best method of holding the rolls in place has proved to be more of a problem than would be supposed after the solution has been found. These rolls could, of course, be riveted to the conical surface, but the rivets would be quite small and rather frail for everyday service, and this arrangement would not provide the lateral adjustment that is desirable to compensate for minute variations in the gear teeth, due to cutting or hardening.

SOME SOURCES OF LEAKAGE IN THE ENGINEERING INDUSTRY

IMPORTANCE OF EFFICIENT TOOL EQUIPMENT, LAY-OUT OF PLANT, ADAPTABILITY OF DESIGN, AND AMOUNT OF STOCK

BY A. A. PEEBLES*

It is impossible in the scope of a brief survey of the subject to treat adequately all the different sources of leakage which may arise in the running of an engineering plant. Each individual case which is examined has its own leakages, some of which may be peculiar to itself, and due to an inherent fault of original planning or to a defect in the management or discipline of the organization. Yet there are some phases of the question under consideration so common, but so seldom emphasized, that a brief summary of the various leakages most frequently met with may not be out of place.

For the purposes of this article leakage may be defined as all unproductive expenditure of time or effort on the part of either the men or the machinery employed in an industrial undertaking, and may also embrace the unprofitable expenditure of capital. It is, moreover, one of the most prevalent and insidious diseases which attack efficiency—prevalent because there are so very few of even the most modern plants into which leakage in one form or another does not creep; insidious because those who suffer from it so rarely know of its presence. To become aware of a source of leakage is frequently to overcome that leakage. It is the loss which is unnoticed that eats into efficiency and consumes dividends.

There are not a few engineers—owners or managers of engineering plants of one kind or another—who blame competition for the loss of business and the decrease in profits which their organizations experience. To a certain extent this may be the cause, but only to a certain extent. If a business loses contracts because other firms underbid its quotations, or if it can secure contracts only by quoting prices which permit of no profit, it is a sure sign that other firms can do the work more cheaply. They are out for profit also, and under normal conditions will not quote prices that do not yield a fair return. Any firm which cannot compete with such prices is operating inefficiently. In other words, the aggregate of its leakages is high, for efficiency is merely the absence of leakage.

It is, perhaps, no exaggeration to say that of the total money expended in the engineering workshops of the world, one-fourth is wasted on unproductive effort of one kind or another, and that of this fourth at least a half is conservable. Such items as loss of time and power, and the wear and tear of machinery involved in the return stroke of slotting, planing and shaping machines are examples of wastage which cannot well be eliminated at the present stage of machine tool development, although the day is not far distant, to judge from present indications, when even such losses as these will be avoidable. It is our aim, however, to deal here only with such of the more common leakages as are at the present moment wholly or partially remediable.

Unsuitable and Poor Equipment

One of the most prevalent sources of leakage in the engineering works is poor equipment. Too much stress cannot be placed upon the importance of up-to-date machine tool equipment. Any efficiency expert in the engineering field will endorse the statement that old or unsuitable tools are responsible for more inefficiency than is any other cause. Factory managers and owners seem to be reluctant to invest in a new plant while the old can, by any means, be made to suffice. When a machine is worn out they will scrap it, but very few will scrap a comparatively new tool merely to make room for a more efficient one. Yet, in deciding when to throw out an existing machine to make room for a new one, the question of the condition of the former should not be the determining factor. The question should be looked on purely as one of business. The operating cost of the existing machine should be computed, and also the operating cost of the proposed new one. The output of each should be carefully estimated, and the final decision based purely upon the results thus obtained. If the installation of the new tool is going to return a reasonable interest on the capital invested, the new tool should be put in. In estimating operating expenses, such items as a percentage of rental and rates proportional to the floor space occupied by the tool, and also a percentage of the interest on the capital invested in the building and of the depreciation thereof, should be considered in addition to such items as interest on capital invested in the tool itself, depreciation, operator's wages, and power used.

The only occasion upon which there is room to question the advisability of installing a more efficient machine tool in place of one of smaller capacity is when the demands of the plant are such that they can be conveniently coped with by the existing machine. In such a case the only saving effected by a change is in the operator's time, the overhead charges of interest, depreciation and floor-space, rental being the same in each case. Such instances are, however, rare, and in most cases the progressive policy of scrapping out-of-date equipment for that of a more modern nature will be found to pay.

The writer recalls one instance which illustrates the effects of wholesale scrapping. In the north of England there is a small general engineering and millwrighting shop which, until a few years ago, was still operating an equipment composed of old, slow-speed tools. It was a busy shop, and in order to do all the work considerable overtime was necessary. It never paid. Work had to be taken at prices that would not permit of the extra cost of overtime in addition to the high cost of production due to inefficient equipment.

A change of management, however, put things on an entirely different basis. The new man raised money and put in a practically new equipment consisting of powerful high-speed tools. He also installed a new power plant with electrical distribution, and speeded things up all around. The result was that the capacity of the plant was augmented to such an extent that its output was materially increased without the necessity of any overtime, and the cost of production was so reduced that the firm was able to quote lower prices than formerly and realize a useful profit in the bargain.

Instances parallel to the foregoing could be multiplied at will. There are many shops operating at a low efficiency simply because the management is reluctant to throw out machines until they are worn out. It is comparatively rare to find an instance where the reverse is the case. That such a contingency is possible, however, will be illustrated by another experience. In this case the shop (also a small one) specialized in valves and steam fittings. Among other machinery was a battery of three hollow spindle capstan lathes which were employed in turning out spindles for a patent valve from the bar. These machines were quite capable of meeting current requirements and were rarely required to do overtime work. It happened, however, that while visiting a machinery exhibition the manager saw some new machines of the same type, but of a more powerful and improved design, and was so impressed by them that he ordered three and had them put in place of the old ones. The approximate net cost of the change was $3500. The saving effected was the money previously paid in overtime, which rarely exceeded $10 a month, and was usually less. This represents a saving of considerably less than 3½ per cent—a poor investment from a business point of view. Yet against this it must not be forgotten that the maximum output of the plant had been increased against possible future demands.

Inefficient Power Generation

Apart from the machine tool equipment of an engineering works, considerable loss may arise from inefficient power generation and distribution. It has been estimated that the average cost of power in the different industries amounts to only 5 per cent of the total cost of production, and for this reason it is sometimes regarded as immaterial. Yet in these days of keen competition, even the smallest source of leakage is of importance, and in the progressive shop, obsolete and inefficient power arrangements are no more to be tolerated than obsolete machine tools. The writer knows of

* Address: Box 1514, Winnipeg, Manitoba, Canada.

two cases which show what can be done in the way of improving power facilities. Both are extreme, perhaps, and consist in replacing a plant of exceedingly wasteful nature by the most efficient available. In one instance the saving in power paid the whole cost of the work of installing the new plant in three years. In the other all the machinery of the factory was driven, and the building lighted with electricity, for a cost approximately equal to that of lighting the place with town gas, as had been done previously.

The lay-out of a plant is a very important factor in the equation of efficiency. This should be carried out on lines calculated to reduce the amount of handling to a minimum. Repeated and unnecessary handling of material during the process of manufacture is a potent cause of costly production and one which is very frequently found. Material should be made to pass, in so far as is possible, through the different departments in order. The disposition of traveling cranes is an important consideration, as is also the method in which the different classes of machines are grouped according to their purposes and in relation to the other parts of the plant.

Where an engineering works has been originally laid out along lines which are not the best, it may not always be advisable to go to the heavy expense of entirely remodeling the whole plant. Yet much can be done in many directions to reduce the unnecessary handling, if a little thought be given to the matter, without undue expense. An example of this is afforded by a large Tyne-side shipyard in the "old country." Two cupolas were operated in connection with the foundry and were kept constantly busy. At one time these cupolas were fed by means of hydraulic lifts. Fuel and pig iron were loaded from the cars on a neighboring siding onto the lifts and were then fed to the cupola—two handlings only being required, but when the quantity of material is considered the second handling constitutes an item of some importance. By building a trestle incline from grade level to that of the charging platform, and running the cars up directly to the latter, the second handling was obviated at a relatively small expense. The direct saving effected in laborers' wages was considerable and, in addition to this, the time saved in charging increased the efficiency of the whole foundry.

Up-to-date equipment in the way of machine tool fixtures and accessories, special tools, jigs and small tools, is scarcely less important than the efficiency of the tools themselves. A good tool-room foreman and staff are among the most valuable assets of the modern engineering works. To have a complete and well kept stock of tools and accessories stored in such a way that any particular one can be found at the shortest notice is to increase the efficiency of the whole plant. Good store-keeping is also an important factor and the reverse a prolific source of small leakage. When men have to wait several minutes while the storekeeper is looking for a certain tool or some stock material, there may be a considerable waste of time, particularly when such waiting is a general thing. It is important that material, both in the tool and stock stores, should be so disposed that anything asked for can be produced without delay.

Accumulation of Scrap

There is another source of leakage which, though usually small, is found in a large number of engineering establishments and is rarely reckoned with. The accumulation of scrap is a source of small leakage in nine out of every ten shops and it is one which is most easily overcome. Scrap in a shop or in a yard takes up valuable space; it is in the way and is generally a nuisance. It also represents so much capital lying absolutely idle. It has a cash value and its value, if realized, might be used to advantage. In the majority of engineering works, far too much scrap is allowed to accumulate before it is disposed of or put to useful purposes in the cupola. The writer was once called on to make a valuation of the entire plant of one of the largest ship-building concerns in the north of England. The estimated weight of the scrap which littered the yards and the different departments was 1000 tons, much of which had obviously been lying around for years. Cast iron, steel and wrought iron were all piled together, with a considerable proportion of brass and gun metal in the form of bearing bushes, stuffing-boxes and the cylinders of old feed pumps. Our estimated value of the whole of the scrap was $21,000, which sum, invested at the rate of five per cent, would give an annual income of $1050. A small item, perhaps, in view of the size of the plant, but one which counted. It was a dead and unnecessary loss, and in addition to this, the scrap was in the way and was occupying valuable space.

The leakages previously dealt with have been those most commonly met with in the operation of the works department of an engineering firm. There are others which have their origin in the offices, and which are not without their importance in the matter of efficiency. Among the chief of these is the matter of adaptability in the designing department. Let us consider a firm that manufactures steam fittings on a large scale as an example. In such a firm it will often be found that every valve or other fitting is designed with no regard for existing types, or for types undergoing contemporaneous design. The result is that for every type of valve or other fitting, an entirely new set of details has to be made. By a little adaptability in design it would be possible very often to utilize, in new designs, existing patterns for the main castings and existing types of spindles, seating and other details. The saving effected in manufacture would be considerable if in the new design the same patterns used in an existing design could be employed. This principle applies with more or less force in many other lines of manufacture. The secret of the success of the manufacturers of the Ford automobile has been duplication—the making of all their engines alike. This has rendered possible the production of an efficient car at a price unheard of previously.

There is still one other point to touch on in connection with the executive end of the engineering business before this cursory review of the matter of leakage is brought to a conclusion. That is concerning the matter of stock. From the engineer's point of view stock can be divided into raw material and finished product, and a careful adjustment of both these divisions is essential to efficiency. In buying raw material, under which head may be included pig and bar iron, and often studs, bolts and other material not produced in the works, the markets must be carefully watched, and purchases made in so far as is possible when the market is favorable. Sufficient stock must be carried in all lines to avoid the possibility of running short unexpectedly, with its attendant holding up of work and waste of time; on the other hand, too much stock should not be purchased. Stock in the shop is capital idle, and it takes up space, which is also true of finished stock. A sufficient quantity should be kept on hand to meet possible demands, and to guard against loss of business on account of inability to give early delivery, but an excess of stock above this quantity is inadvisable. It represents an unprofitable investment of capital, and there is the risk that some of it may become out-of-date before it is disposed of. This adjustment of stock, both in regard to raw material and finished product, is one which demands nice judgment. In quiet times manufacturers not infrequently continue to run their plants at or near capacity, with the result that a surplus stock is accumulated which is difficult to dispose of. Stock should be kept within reasonable limits even if it means a shut-down for a time. It is the best policy in the end.

* * *

An interesting application of the vacuum cleaner as a foundry tool is told by a contributor to *Foundry*. When making a three-part mold, six spikes were inserted to secure the sand, but one spike passed through the sand into the pattern, and after rolling over and drawing the main part of the pattern it pulled through the core and about a handful of sand dropped down underneath. In this case, either the mold had to be destroyed or the loose sand had to be removed. The only feasible manner in which the latter could be done was by the use of a vacuum cleaner, by means of which the loose sand was readily sucked from underneath the core. The resulting casting was sound, which showed that a device of this kind may be utilized to advantage in foundry work.

DESIGN AND CONSTRUCTION OF BORING TOOLS

VARIOUS TYPES OF BORING TOOLS FOR ENGINE LATHES, HORIZONTAL AND VERTICAL TURRET LATHES OR BORING MILLS

BY ALBERT A. DOWD*

A boring tool or boring-bar is, in itself, a very simple tool and yet, in its various applications, it may require considerable forethought in order to obtain a tool which will be exactly the right one for the job. In order to properly design any kind of a cutting tool, an intimate knowledge of the actual working conditions which are met with in using the tool is a valuable asset. There are a number of factors which influence the design of boring tools and there are also many types of machinery to which boring tools may be applied. In some cases the bar revolves with the spindle of the machine, while in others it is held rigidly and the work revolves around it. These things affect the design and must be considered. The work naturally controls the size of the bar and also its shape, while the material which is to be cut makes a difference in the shape of the tool and determines the amount of "chip clearance" necessary.

The tools described and illustrated in this article must be considered as representative types of the many varieties to be met with in the general course of manufacturing. Points in design and construction will be noted and faulty tools will be discussed and criticised.

General Points in Boring Tool Design

Some of the important points in the design and construction of tools for single-point boring are here given, and while some of these may seem obvious, they may be of assistance in calling attention to matters which would otherwise be overlooked.

1. Chip clearance must be very carefully looked after when the tool is to be used for cutting steel, as an accumulation of chips caused by insufficient clearance is very annoying to the operator and also injures the work by tearing or scratching it, and finally ruins the bar itself unless it is hardened. The amount of clearance between the bar and the work should be as great as possible without sacrificing strength, and in this connection it should be noted that in addition to the necessary chip clearance at the point where the cutting action takes place, provision must also be made to get rid of the chips themselves. For this reason the portion of the bar beyond the cutting tool should be so proportioned that chips will not wedge. In cutting materials other than steel the clearance is not so important, as the chips are short and do not curl up or cling to the bar, so that they practically take care of themselves.

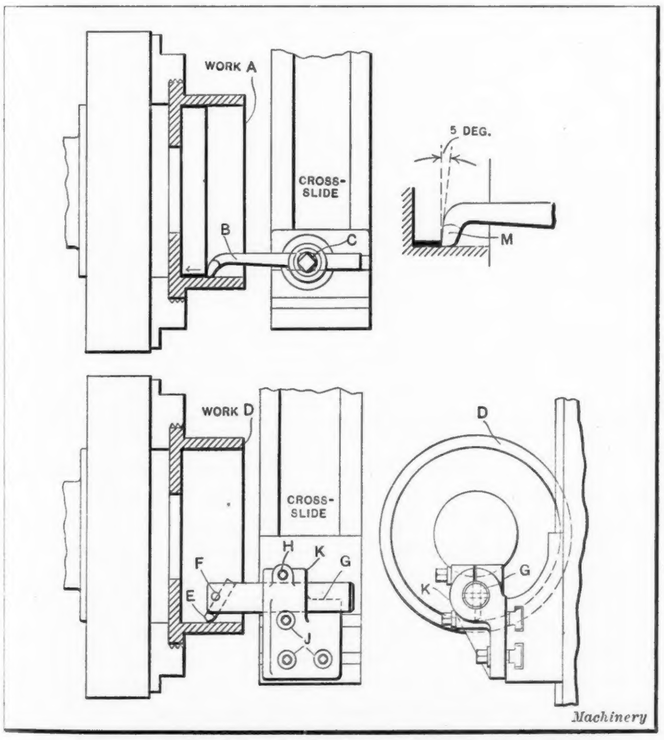

Fig. 1. (Upper View) Forged Type of Boring Tool; (Lower View) Boring Tool with Inserted Cutter

2. The method of holding or clamping the tool in position should be such that the thrust of the cut comes against the solid body of the bar and not against the set-screws or clamps. It is advisable to use square-head set-screws instead of the headless type whenever possible.

3. Boring-bars should be provided with some means of adjusting the tools for diameters, by the use of "backing-up" screws or wedges. The so-called "sledge hammer adjustment" type of bar should never be used when there is room enough to put in adjusting screws.

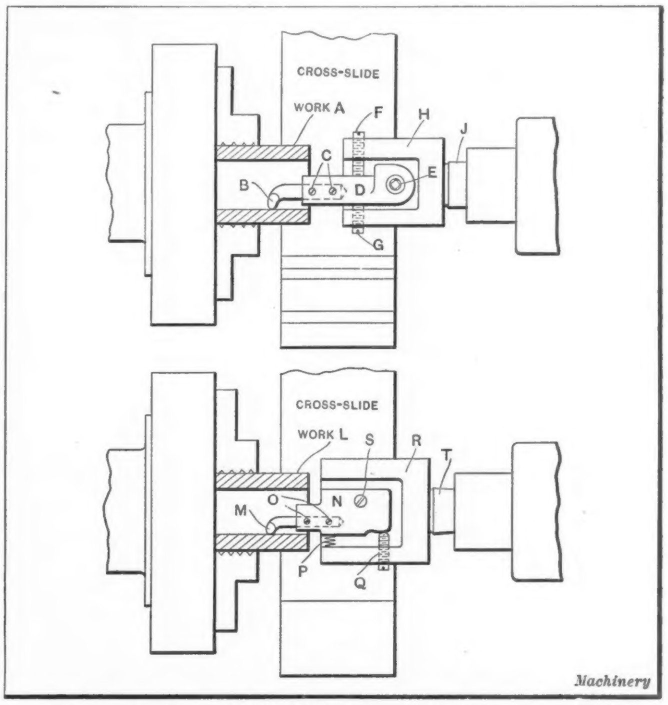

Fig. 2. Two Types of Adjustable Boring Tools for Tool-room Work

4. Boring-bar tools should be made as large as the diameter of the bar will permit without sacrificing strength, in order to assist in carrying away the heat generated by the cutting action, and to permit the use of heavier feeds without burning the tool. The rake of the tool should be such that it will turn the chips to the best advantage.

5. The bar should be so designed that micrometers can be used over the bar and tool in order that the operator may be able to set his diameters closely without resorting to the usual "cut-and-try" method used by our forefathers.

6. In the design of multiple boring-bars which are to be used to bore up to a shoulder, it is not good practice to set the tools in the bar at an angle. They should be located in a plane perpendicular to the axis of the bar. If set at an angle it will be found a very difficult matter to grind the tools so that diameters and shoulder distances will remain constant.

7. Bars designed for use on turret lathes should have the tools set in a plane perpendicular to the rotation of the turret. By this means variations in the indexing of the turret are minimized in their relation to the cutting tools, so that diameters can be held much closer to size than if the tools are arranged in a plane parallel to the turret rotation.

8. When the work is of such a nature that a cutting lubricant is required, provision should be made so that an ample supply of the fluid can be carried directly onto the face of the cutting tool. This result can be accomplished either by means of a hole in the bar with outlets at the proper places, or oil grooves covered with a strip of sheet brass. In either case a good connection must be made with the cutting lubri-

* Address: 84 Washington Terrace, Bridgeport, Conn.

cant system on the machine. This may be arranged by a distributing collar on the turret or by means of a special oiling device through the spindle.

Boring Tools for the Engine Lathe

Boring tools which are designed for use in the engine lathe are generally of a very simple kind, adapted only to light cutting and seldom used for more than one or two pieces of

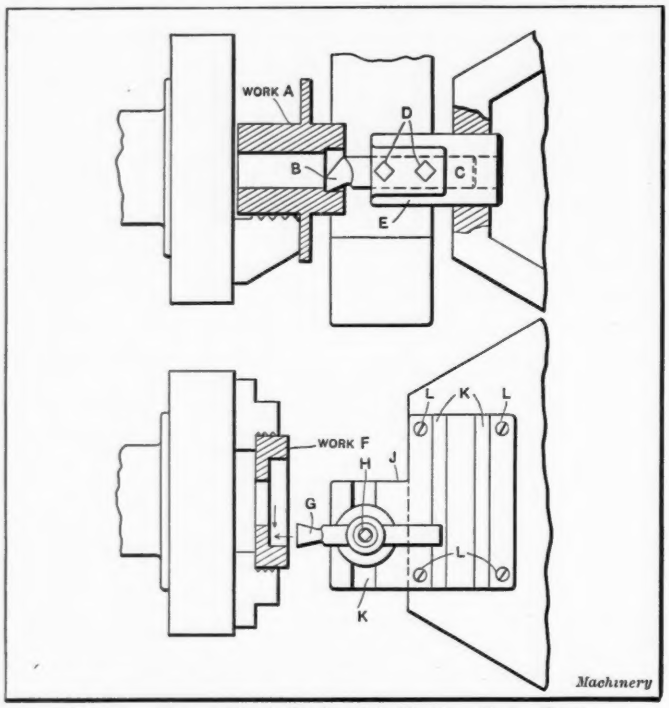

Fig. 3. (Upper View) Single-point Starting Tool; (Lower View) Boring and Facing Tool

work of the same size at the same time. Several varieties are to be found in the average tool-room, although forged tools will be noted in greater numbers than any of the others. Tools of this kind of almost every conceivable shape and size, from a small round "hook tool" for cutting an inside recess, to a a large bar of tool steel bent over at the end for boring some long pieces of work, will be found in abundance. There are

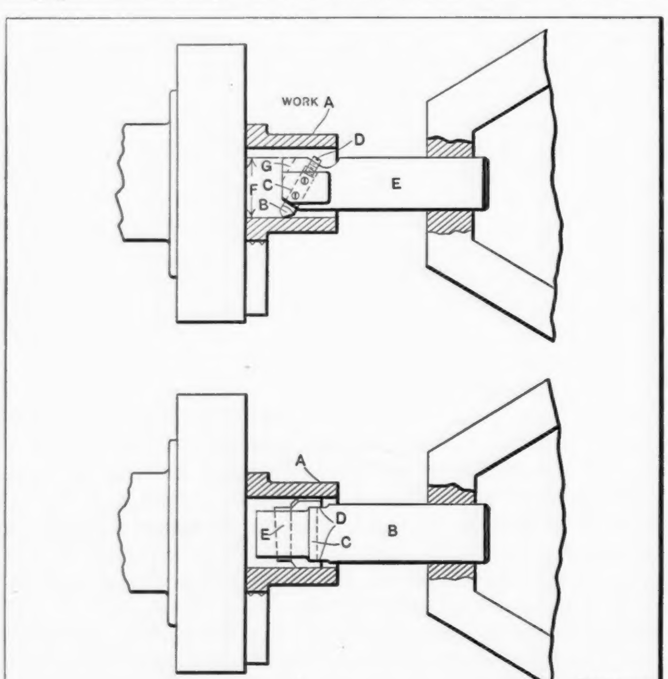

Fig. 4. (Upper View) Boring-bar with Adjustable Cutter; (Lower View) Boring-bar with Double-ended Cutter

square bars and round bars with inserted tools, and, in addition to these, each toolmaker has a special boring tool of his own make which he uses for jig work. These special tools occasionally show considerable ingenuity in their construction, and are usually made in such a way that very fine adjustments can be attained.

The upper part of Fig. 1 shows a piece of work *A* held by the outside in chuck jaws, the machine on which the work is to be done being an engine lathe. A plain forged tool *B* is held in the toolpost *C* on the cross-slide of the lathe. This type of tool is the simplest of all tools used for boring and consists of a rectangular piece of tool steel of suitable size to fit the toolpost. The tool is drawn out and bent over at the cutting end by the blacksmith and is then ground to a cutting edge by the workman using it. Hundreds of tools of this variety can be found in every machine shop and factory in this country. They are suitable only for light cutting and there is a tendency toward "chatter" even when the cut is light; this is due partly to the shape of the cutting end and partly to the overhang of the entire tool. It will be found that less chatter will result if a slight land or flat is stoned on the tool immediately below the cutting edge. The tool should also be set slightly above the center. For casting work where scale is encountered, there is a decided tendency for the tool to ride up on the scale and ruin very rapidly if it is ground as shown at *B*. The enlarged view *M* shows another method of grinding which is useful in cases of this sort. It will be noted that there is a slight back taper to the end of the tool and this assists in preventing any riding up on the scale, as its tendency is to make the cutting point draw in slightly and thus keep under the scale. Care must

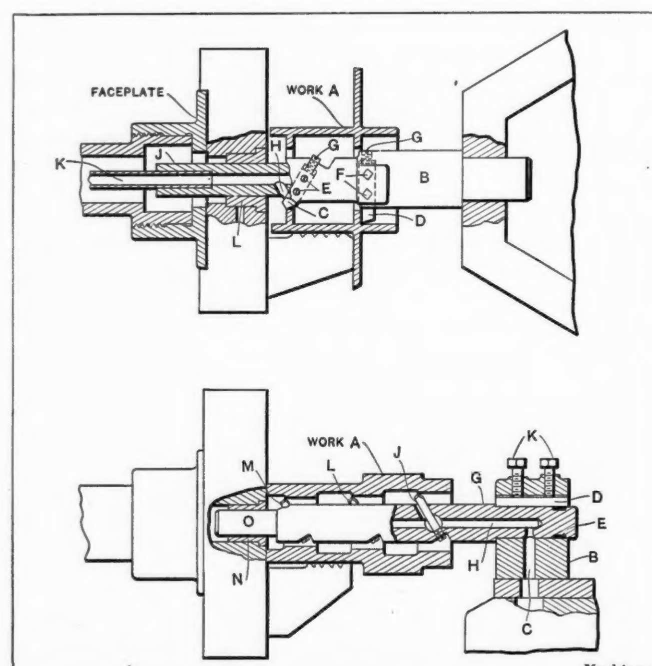

Fig. 5. Boring-bars arranged for lubricating Cutters

be taken not to make the angle too great—5 degrees is ample, and much less than this can be used if desired.

The lower part of Fig. 1 shows the same piece of work *D* with another type of boring tool in action. A cast-iron body *K* is held down on the cross-slide of the lathe by means of the three bolts *J*. A steel bar *G* is longitudinally adjustable in the cylindrical portion of the holder and is clamped in position by means of the binder screw *H*. A round cutting tool *E* is held in place by the taper pin *F*, in a manner familiar to all. A holder of this type will be found a very useful adjunct to any toolroom, and is adaptable to a variety of conditions. A series of bushings can be made to take different diameters of round stock, and tools may be quickly made to suit almost any case. Obviously, adjustment for diameters is made by the cross-slide. Rigidity and adaptability are points in favor of this device.

Adjustable Boring Tools for Jig Work

Fig. 2 represents two styles of adjustable boring tools used mostly for boring small shallow holes or jig bushings. These tools are capable of fine adjustments but are not suited for any kind of heavy cutting. The upper part of the illustration shows how tool *B* is used for boring a part of the bushing *A*, which is held in chuck jaws. The body of the toolholder *H* is made of steel and is turned down and tapered at

J to fit the tailstock spindle. The adjustable swivel *D* is pivoted on the shouldered screw *E*, and is adjusted by the two headless set-screws *F* and *G*. The tool *B* is of round section and fits the end of the swivel, where it is held in place by the two screws *C*. The end of the tool is bent over for the purpose of clearance. A tool of this kind is very convenient and is easily adjusted for diameters within its capacity. It is not adapted to deep holes, but can be made up in several sizes so that it will handle fairly large work.

The lower part of the illustration shows another tool of somewhat similar construction, which is designed for the same purpose as the other. The body is of steel and the shank *T* is turned taper to fit the tailstock spindle. The forward portion of the body *R* is cut out to receive the swivel *N*, which pivots on the screw *S*. The tool *M* is of round section, bent over at the end, and it is held in place by the two screws *O*. One adjusting screw *Q* is all that is required in this tool, as the coil spring *P* takes up lost motion and prevents drawing in. This tool is not as rigid as the one previously referred to, but the spring makes adjusting much quicker, as only one screw is needed. A number of tools of this type, and of various sizes, were made for tool-room use in a large automobile factory and were used on the greater part of the jig work.

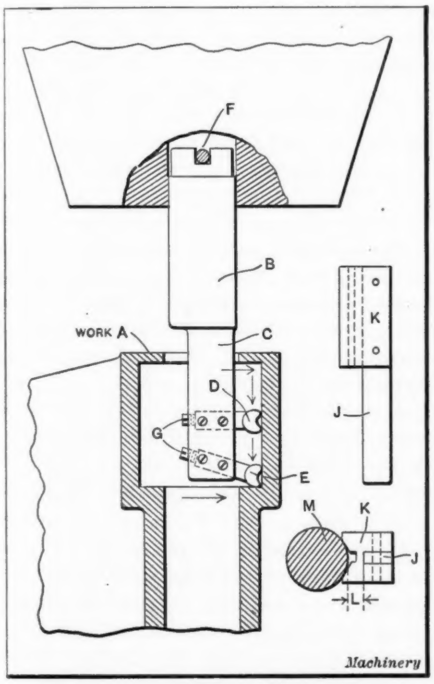

Fig. 6. Bar for Undercutting, Facing and Boring in the Vertical Turret Lathe and Gage used in setting Tools

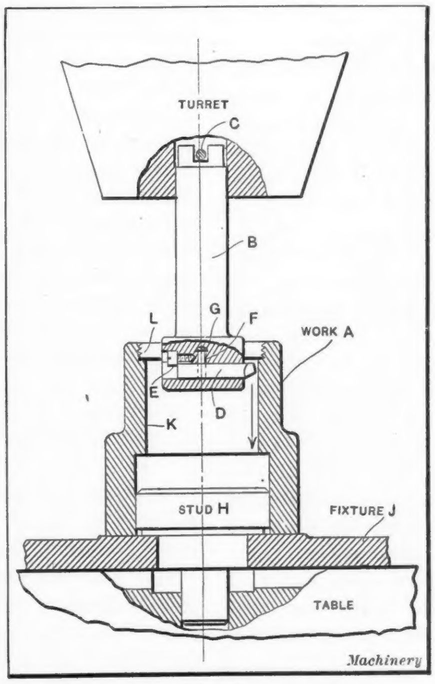

Fig. 7. Bar equipped with a Set of Interchangeable Cutters for Boring and Reaming

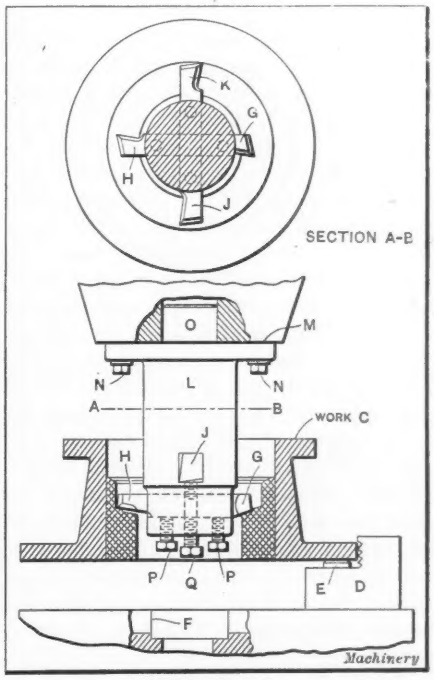

Fig. 8. Bar with Four Cutters set at Different Radii for removing Considerable Stock in One Cut

Boring Tools for the Horizontal Turret Lathe

Boring tools which are required for use on the horizontal turret lathe are of many forms and their design is somewhat dependent on the type of machine to which they are to be attached. On machines having no transverse movement to the turret slide, the tools are nearly always designed for straight boring, while on the other types of machines, *i. e.*, those having transverse movement, the design is frequently made in such a way that the tools can also be used for facing operations. The form of the turret itself also influences the design to a certain extent, for it is evident that a flat turret would require a different type of tool-holder than one of the vertical face variety.

Single-point Starting Tool for Taper Holes

The work *A* shown in Fig. 3 is a malleable iron automobile hub with a cored taper hole which runs out of truth very badly. Therefore it was necessary to design a starting tool of the single-point variety in order to generate a true running hole, so that the subsequent tool would start properly without being influenced by the wobble of the core. This tool and tool-holder are very simple, the tool itself being a piece of round high-speed steel bent over on the end and ground to cut a diameter a trifle smaller than the large end of the tapered hole. The holder *E* is a piece of machine steel of cylindrical shape, which is ground on the outside to fit the turret hole and on the inside to fit the shank of the tool *B*. Two set-screws *D* are used to hold the tool in position. It will be noted that the end of the cutting point is ground very nearly square so that it will not ride up on the scale. The tool is not made for continuous boring but is merely used to generate a true hole for a short distance into the cored portion of the hub.

Boring and Facing Tool for a Flat Turret

An example of a boring tool which is also used for facing out a pocket on a turret lathe having a flat turret is shown in the lower part of Fig. 3. This tool is of the "shovel nose" type and its cutting action is rather hard on account of the bluntness of the nose and the amount of stock which it removes. The work *F* is a machine steel forging and the shoulder is not recessed at all in the blank. The tool *G* is of rectangular section and it is forged and ground on the cutting end to the shape shown. The tool-holder *H* is supported by the steel bracket *J* which is fastened down on the turret face by screws *L*. The slots *K* are T-shaped and permit various settings and combinations of tools to be made.

Fig. 4 shows a very simple type of single-point adjustable boring-bar for machining the bushing *A* (see upper part of the illustration). Although this bar is simple in its construction, there are several important points in design which should be carefully noted. The bar *E* is of a low grade of tool steel and is hardened and ground to fit the turret hole. The reason for making the bar of tool steel is simply to obtain all the rigidity possible and thereby obviate chatter. The tool *B* is of round section and is put through the bar at a slight angle, being held in position by the two screws *C*. A backing-up screw *D* permits careful adjustment to be made. The bar is cut away where the tool comes through to provide chip clearance, but it is cylindrical on the end except in this one place. By making it this way, it is found an easy matter to use micrometer calipers across the bar and tool as indicated at *F*, so that accurate settings may be readily made without resorting to "cut-and-try" methods. It is very bad practice to bevel the end of the bar at *G* and put the holding screws through at this point, because a calipering point is sacrificed by so doing. A simple formula is here given for setting tools of this type for turning a given diameter:

Let F = required calipering distance for a given size hole;
X = diameter of the bar at the end where the tool is;
Y = diameter of the hole to be bored.

Then

$$F = \frac{X + Y}{2}$$

This formula will be found useful for setting tools very close to the desired diameter, although the spring of the bar will cause slight variations and the amount of stock which is to be removed also makes some difference.

The lower part of Fig. 4 shows a boring tool of an entirely different type. The cutter is double-ended, and a bar of this sort is often used for removing stock rapidly. Although it is a faster cutting tool than a bar having only a single tool or cutting point, it cannot be depended upon to produce a hole which is absolutely concentric with other surfaces machined at the same setting. The work *A* is the same as in the upper part of the illustration, and the bar *B* fits the turret hole. It is flattened slightly on two sides at points *D*, and a rectangular slot contains the cutter *C* and the wedge *E*. It will be noted that the cutter is shouldered so that it is a close fit at the points *D*. Tools of this type cannot be ground radially without changing their diameters, but this is seldom necessary as the cutting edge is at the forward end. A land of about 1/8 inch is usually left just back of the bevel, and the cutter can be ground back to this point without changing the diameter. Beyond this, however, there is a slight back taper for the sake of clearance, so that the life of the cutter does not extend beyond it.

Boring-bar with Provision for Cutting Lubricant

On certain classes of work it is very difficult to supply the cutting points of the tools with sufficient lubrication to make them thoroughly efficient, when the regular supply system is used. Some method must be devised, therefore, to direct the flow to the point or points where the cutting action takes place. An example of a bar arranged to carry the lubricant to inaccessible tools is shown in the upper part of Fig. 5. The work *A* in the chuck jaws is an automobile hub of malleable iron. It will be noted that the portion bored by the forward tool *C* is in such a position that it cannot be reached for lubricating purposes in the ordinary way, but the rear tool *D* can easily be taken care of. The boring-bar *B* is of a low grade of tool steel and fits the turret hole at the rear end; the forward end *J* is a running fit in the chuck bushing *L*. A telescoping oil supply tube *K* enters this end of the bar and is supplied with lubricant from the rear end of the spindle. The hole in the bar at *H* leads the fluid directly onto the face of the cutting tool *C*, thus insuring constant lubrication at this point. The two tools are held in place by the screws *E* and *F*, and they are provided with means of adjustment in the backing-up screws *G*. The writer has used bars of this type in a number of cases with very gratifying results.

Fig. 5 (lower illustration) shows a very different condition, in a multi-cutting boring-bar for generating a series of true holes in the bronze artillery hub *A*. The finished hole is tapered but a starting bar was used in order to prepare the hole properly for the taper tools which followed it, so that they would not be influenced by the irregularities of the cored hole. In this case the turret lathe was one of the flat-turret variety, and provision was made for lubrication through the hole *C* in the turret face. As the turret indexed to the proper position, this hole came directly over another in the slide, which, in turn, was connected with the lubricant pressure supply system, thus allowing the liquid to pass up into the body of the tool-holder. The boring-bar *G* is turned down at the rear end to fit the tool-holder *B*, and has an annular groove *E* which is packed with felt to prevent the escape of lubricant. A shoe *D* is forced down on the bar by the two screws *K* and prevents the bar from turning. The hole *H* in the bar is drilled from the forward end and is tightly plugged so that this end remains closed to prevent the lubricant from passing through. A groove is cut in front of the tools *J*, *M* and *L*, as shown at *J*, and this allows the fluid to flow directly onto the faces of the tools. The end of the bar is piloted at *O* in the bushing *N* which is fixed in the chuck body. An arrangement of this sort has also proved successful in a number of instances.

Bar for Undercutting, Facing and Boring in the Vertical Turret Lathe

A very difficult condition for which to design tools is shown in Fig. 6, as the work itself requires rapidity of handling and is a steel casting weighing about 300 pounds. Only a part of the piece is shown at *A*, but it will readily be seen that it is necessary to make the bar in such a way that the tools can be used to do all the cutting indicated by the arrows; *i. e.*, undercut the upper flange, double-bore the interior, and face the lower shoulder. As the fixture on which the work was held was of the indexing variety and was very much off center, it was not expedient to run at high speed. Therefore, the double boring was of assistance in increasing the production. It will be noted that the hole through which the tools pass is of small diameter, which makes the problem still more difficult. The shank of the bar *B* fits the turret hole at its upper end and is slotted so that the pin *F* in the turret will act as a driver. (This feature is patented by the Bullard Machine Tool Co.) The lower part of the bar is eccentric to the shank in order to obtain the necessary clearance when the tools are in action. Even the tools themselves are considerably out of the ordinary in that they will cut in two directions. The upper tool *D* is used for undercutting the flange and also for boring, while the lower tool *E* is used for facing the lower shoulder and partially boring the interior. Both these tools have backing-up screws *G* and are held in place by the headless set-screws.

As it was necessary to set these two tools so that they would cut approximately the same diameter, the gage shown at the right of the figure was made to assist in the setting. The V-block *K* was slotted to receive the steel strip *J* so that distance *L* would measure the correct distance from the bar shown in section at *M*. It is obvious that the gage could be placed against the bar so that tools could be set out the right amount by means of the backing-up screws. This bar gave fairly satisfactory results although some trouble was experienced with chips, as there was considerable stock to remove. There was likewise a slight tendency to chatter when using a heavy feed, but this was remedied by careful grinding to make the cut as easy as possible. It must be remembered that the conditions were about as awkward as they could be, and the lack of room made it necessary to cut down the bar to such an extent that it was hardly heavy enough for the work. Taken as a whole, however, the action was satisfactory for a roughing tool. It was not used for finishing cuts.

Slip-cutter Bar for the Vertical Turret Lathe

The steel hub shown at *A* in Fig. 7 is held on a special fixture, located by the previously bored and reamed hole which fits the stud *H*. The hole *K* has been rough-bored in the first operation, but enough stock has been left for the final finishing so that it may be finish-bored and reamed and part *L* threaded at the same setting. This type of bar is the product of the Bullard Machine Tool Co., and is designed especially for use in their machines. It is a combination boring- and reaming-bar, and the cutters are of the "slip" variety. One bar can be furnished with a set of cutters for the various sizes of holes within its capacity. A full set of cutters for any one size of hole consists of chamfering, rough-boring, finish-boring, rough-reaming and finish-reaming tools. The first three of these are of square section, carefully ground to fit the broached hole *D*. The rear ends of these tools bring up against the shoulder of the screw *E*, which acts as a stop. The fit in the hole is such that tools can easily be put in and taken out with the fingers.

The two reaming cutters are used in the rectangular hole *F* which is at right angles to the other hole; these tools are allowed to float so that they follow the hole generated by the boring tools. The action against the reaming cutters is in an upward direction, and comes against the hardened steel plug *G* which is inserted in the bar. The bar *B* is of special steel and is slotted at the upper end to fit the driving pin *C* which is located in the turret. Bars of this type have a number of advantages, one of which is that only one turret hole is occupied; other advantages are the cost of maintenance, and the adaptability of the bar with its series of cutters to handle a number of different sized holes. The cost of large sizes of reamers of the fluted type is considerable, while a flat reamer such as that used in this bar is inexpensive. It may be noted that the pressure or thrust of the cut is all

that holds the boring tools in place, so that trouble would be experienced if a cored pocket were to be encountered. This is provided for by a detent screw in the end of the bar, which prevents the tools from coming out. This screw can be put in any time if it is found necessary.

Multi-cutting Bar for the Vertical Turret Lathe

An example of a bar designed to remove a large amount of stock in a very short time is given in Fig. 8. The work shown at *C* is a steel boiler nozzle which is forged, and a 5-inch hole punched in it before it is machined by the vertical turret lathe. The finished hole is 8 inches in diameter and it was desired to remove the surplus stock as rapidly as possible. Accuracy in the diameter of the hole was not essential. The bar *L* is a steel forging which fits the turret face at *M* and is held to it by the screws *N*. The stem *O* is used to center the bar in the turret. Two rectangular holes are broached through the body of the bar, at right angles to each other, and these receive the high-speed steel cutters shown. It will be noted that these cutters are so proportioned that they remove the stock in a series of steps, each tool extending beyond the preceding one and also above it about 1/8 inch.

The section taken at *A—B* shows the manner in which the tools extend beyond each other, and the lower view illustrates the cutting action of the tools. One end of the first tool *G* makes the first step, while the other end *H* makes the second. In like manner one end of the upper tool *J* makes the third step while the other end *K* takes out the remainder. The two screws *P* hold the first tool in place, while the other is secured by the screws *Q*. It will be noted that the work is held in jaws *D* and is supported on buttons at *E;* the height above the table is great enough to allow the end of the bar and the set-screws to go through far enough to finish the cut. Regarding the upkeep of these tools, attention is called to the fact that they may be pushed backward or forward to compensate for wear and distribute the cut. For roughing purposes requiring rapid removal of stock and rapid cutting, a bar of this sort has proved very successful, but it is not recommended for work requiring great accuracy.

Other Types of Boring-bars

There are several bars on the market which are adjustable for various diameters by means of micrometer screws and taper wedges. These are useful for many purposes but space will not permit a detailed description nor has it been the writer's purpose to deal with the many varieties but rather with representative types. Special bars for many purposes, porcupine bars and cutter heads of various kinds, I have not

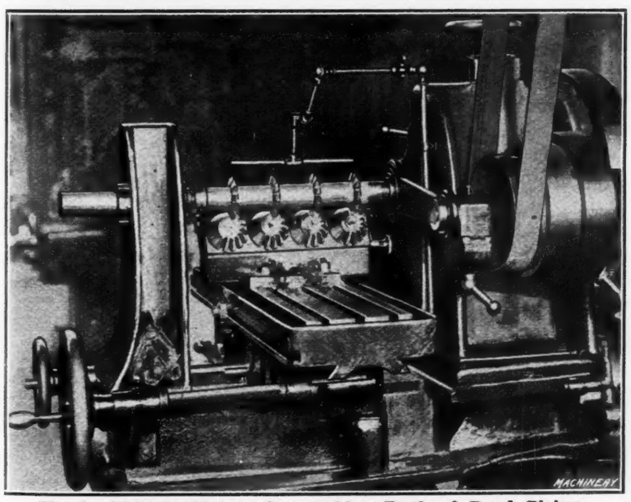

Fig. 1. Multiple Fixture for roughing Teeth of Bevel Pinions, applied to Horizontal Milling Machine

attempted to describe, for these are not single-point boring tools. Neither has mention been made of boring tools such as are used in fixture work on the horizontal boring machine, for these are found in every tool-room, in all shapes and sizes.

In several of the illustrations it may be noted that the tools are shown in a plane parallel with the rotation of the turret. This has been done simply because the details are more clearly apparent when shown in this way. Greater accuracy is obtained by setting the tools in a plane perpendicular to the turret rotation, as previously stated.

MULTIPLE FIXTURE FOR ROUGHING THE TEETH OF BEVEL PINIONS

BY C. BOELLA*

A great many small bevel gear pinions are used as "satellites" in the manufacture of differential gearing for motor cars. In order to increase the production of these pinions and lower the cost, the device shown applied to a horizontal milling machine in Fig. 1 was designed. By using this fixture, gear blanks can be roughed out at a single cut; they are then finished more rapidly by a second operation in a regular bevel gear cutting machine.

The cutter arbor is fitted with four properly spaced cutters. The fixture, which is securely bolted to the milling machine

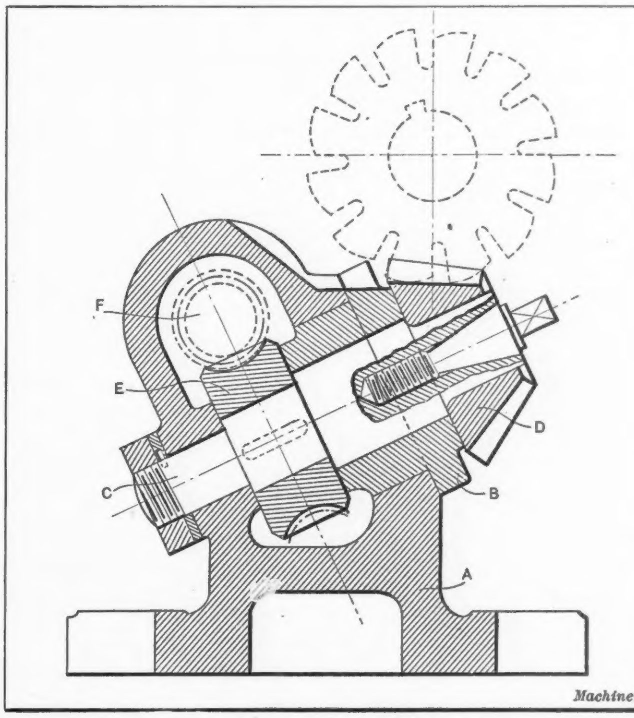

Fig. 2. Cross-section of Multiple Fixture

table, has four small spindles, each holding a bevel pinion at such an angle that the cutters will rough out the teeth at the depth required. When a tooth has been milled in each blank, the table is withdrawn and, by a complete turn of the handle seen on the right side of the fixture, the four pinions are indexed simultaneously one tooth. Then another cut is taken and this operation is repeated until all the teeth are roughed out.

Fig. 2 is a section of the device. There is a hollow base *A* having four plates *B* each fitted with a spindle *C*. The spindles have expansion collets to hold blanks and they are indexed by the worm-wheels *E* which mesh with worms *F* that are turned by the handle previously referred to. The worm-wheels have twenty-eight teeth and the worm is double threaded so that each complete revolution of the worm causes the pinions to rotate 1/14 turn, as they must have fourteen teeth of five diametral pitch. The indexing handle is fitted with a spring knob, the end of which can engage a hole located in the base, thus insuring an exact division. These pinions are made from 5 per cent nickel steel. The time required for each cut is one minute ten seconds, four pinions being completely roughed out in about sixteen minutes. The machine is operated by the same man who runs the gear shaper used for finishing the teeth.

* * *

For an anti-friction metal that can be subjected to pressures up to about 400 pounds per square inch, the *Foundry* recommends the following composition: Lead, 85 per cent; antimony, 10 per cent; and tin, 5 per cent. For pressures exceeding 400 pounds per square inch, the following alloy will prove satisfactory: Tin, 85 per cent; copper, 5 per cent; and antimony, 10 per cent. This alloy can be used safely for pressures up to 1000 pounds per square inch.

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ENGINEERING QUALIFICATIONS

In three months the engineering colleges of the country will be turning out a few thousand graduates to find places for themselves in the industrial world, and many of them with no practical experience are likely to believe that the mere act of graduation opens the road to success for them. We believe that no greater service could be rendered young men fitting themselves for industrial work than for their teachers to impress them, while at college, with the fact that many other qualities are necessary for success in the engineering field besides mere ability to pass their "exams" successfully. The knowledge and training obtained at school and college are important, but there are other qualifications that count for more in later years. Two of these are reliability and thoroughness; another is tact; others of the greatest value are executive ability, push and initiative; while still another is real enthusiasm and love for the work in hand; and, last but not least, health—moral, mental and physical. One engineer, in stating his requirements for subordinates in terms of percentages, said that he would be satisfied with fifty per cent of technical knowledge and analytical ability, provided the remaining fifty per cent was divided into twenty per cent tact and ability to cooperate, and thirty per cent push, initiative and originality. This combination makes for a well balanced personality, to whom success in the engineering field is well-nigh certain.

Reliability in an engineer should not be confused with what is commonly called "honesty," which is merely the inclination to do one's best. Reliability is a faculty which requires constant training, until mere inclination has been developed into ability to get out of oneself the very best possible from one's mental and physical qualifications. One phase of reliability is the ability to carry out definite instructions. This cannot be done without continual training of the mind, and the lack of such training is shown by the inability or unwillingness of many people to carry out a given piece of work according to given instructions. Another evidence of reliability is a willingness to acknowledge promptly one's inability to carry out a certain piece of work in a prescribed way or within a specified time limit. Paradoxical as it may seem, a definite statement of what he *cannot* do may often be the first thing to attract favorable attention to a young man.

GETTING IN TOUCH WITH THE FOREMAN

Some time ago a manufacturing concern made for stock a number of parts consisting of steel castings bushed with a well-known bearing alloy. The shafts were required to make but a short oscillatory movement in the bearings, and the foreman instructed the fitters to give them a clearance of 0.001 to 0.0025 inch, the diameter being 1½ inch. The parts were ordered out for a repair job a few months after and the workmen discovered that the shafts would not enter the bearings. An investigation showed that the diameter of the holes was 0.005 inch less than the shaft diameters.

The foreman was called to account, and while he was positive that the parts had been fitted with sufficient clearance, he was unable to explain satisfactorily why the change had taken place. A letter to the editor describing the case brought an explanation clearing the foreman of blame and saved his job.

The cause of the trouble was that the alloy bushings had been assembled in the steel parts by pressure. The alloy used is one having excellent characteristics as a bearing metal, but though very strong, it "flows" slightly when subjected for a long time to heavy pressure. The result was that the bushings contracted in diameter, causing the closing of the holes and thus making them too small for the shafts that were originally fitted in them.

The makers of the alloy are fully aware of this characteristic, and issue directions to the effect that when used for bushings it should not be assembled by pressure. Instead, it should be made a close shove fit and held in place by non-bottoming set-screws. When properly fitted according to directions, it gives perfect satisfaction.

This brings us to the practical difficulty met with by some concerns in getting their products treated as they must be treated in order to yield the best results. They deal with purchasing agents who, in some cases, having little appreciation of technical matters, are careless in transmitting important specifications to the men in the shop. Manufacturing concerns should find it to advantage to encourage free intercourse between the representatives of concerns furnishing them supplies and the men responsible for results, especially when the treatment is out of the ordinary. The free circulation of catalogues and circulars among the men has been found advantageous. In some places this literature forms part of a regular circulating library. MACHINERY will gladly publish important details of practice pertaining to new products in its field and invites all manufacturers to use its columns freely as an educational means to reach the foremen and machinists handling them.

* * *

THE OBJECT OF TEMPERING STEEL

In examining recently a great number of books and articles on the subject of tempering steel, we found that almost invariably the authors stated that the object of tempering was the softening of the steel. They said that steel, when heated to the required hardening temperature and quenched, became too hard, and that therefore it was necessary to draw the temper in order to soften the steel so that it could be used for the purpose for which it was intended. This is an erroneous statement which seems to be accepted by the majority of mechanics, because they fail to distinguish between hardness and brittleness, two entirely different qualities. Hardened steel is tempered in order to make it less brittle, but unfortunately the tempering process also softens the steel to some extent. If it were possible to temper steel so as to produce greater toughness and at the same time retain the extreme hardness, the ideal condition would be obtained. That hardness and brittleness are not necessarily synonymous may be seen in the case of cast iron, which is very brittle but not very hard. On the other hand, there are some alloy steels that may be made very hard and at the same time very tough. The object of tempering steel is to reduce the brittleness; the hardness is simultaneously reduced, but this unfortunately cannot be avoided.

PUNCH AND SHEAR FRAMES

The article "Punching Machine Frames," to be found in another part of this number, is one of the most interesting contributions of a practical nature that has ever been published on this important subject. The author shows the danger of using the commonly accepted method for calculating the design of machine frames of the curved type, and gives a method which can be used without difficulty by any one familiar with the rudiments of the use of formulas. In the deduction of the formulas, the author has made use of calculus, but it is by no means necessary to be versed in calculus in order to appreciate the important facts brought out and profit by the results of the calculations.

Probably many readers of technical journals are deterred from reading certain articles because they find that they contain mathematical treatment of a kind with which they are not familiar. As a rule, this is a mistaken idea, because even if the reader is not able to follow the writer's mathematical proof of his theories, he can nevertheless profit by digesting the results obtained, taking for granted the accuracy of the writer's deductions. Of course the value is still greater to the man who can follow the author through the mathematical treatment, but this is not necessary to understand the article.

In the specific case referred to, the fact that the stresses in machine frames, when calculated according to the more modern theories evolved, may be from two to three times the stresses found by the older methods of calculation, is sufficient to indicate the importance of the subject. It is true that accurate analytical methods that can be easily applied have not as yet been developed, but the method explained in the article referred to is so nearly correct in the light of modern investigations that it might well be adopted as a standard for curved machine frame calculations until a better and simpler method giving the same accuracy is proposed.

* * *

THE SMALL POWER PLANT

The power problem confronts every factory manager in one form or another. Small shops in cities can *sometimes* buy electric power cheaper than it can be generated with small power units, but unfortunately this is not always true. The fact that the large power companies cannot or will not uniformly furnish electric power at rates cheaper than it can be supplied by small private plants is a standing reproach to the good sense and business management of these so-called public utilities. Manifestly, the large power plant managed on scientific principles, generating steam in boilers fired with the cheapest coal and utilizing it in turbines of great size and high efficiency, can produce a kilowatt at the plant at a far lower price than can the small concern. The overhead charges are less proportionately and the efficiency of production much higher. Many manufacturing plants have in self-defence put in power plants in order to get reasonable rates from the electric companies. It becomes an idle investment but a necessary one, if reasonable rates are to be had.

But aside from urban conditions are those where power must be generated at the shop with small units. What shall the prime mover be? The steam engine and boiler is expensive to maintain. A fireman and engineer must be in constant attendance. A gas engine using illuminating gas is likely to make big gas bills. Small gas producers are successful, but require more technical knowledge to operate satisfactorily than is available many times. The gasoline engine is out of the question now with gasoline at twenty to twenty-five cents a gallon. The kerosene engine is far more economical, but of yet greater economy is the Diesel engine. This motor burns fuel oil, distillates and other cheap liquid fuels. The engine is simple in construction, but must be well built to give long service. It is highly economical, the thermal efficiency running above twenty-five per cent and the fuel consumption being as low as one-half pound of oil per indicated horsepower. The high compression necessary in the Diesel engine is the greatest disadvantage. It makes necessary a power starting device for even the smallest units, but compressed air furnishes the means for it without serious complications.

THE ADVANTAGES OF MANUAL TRAINING HIGH SCHOOLS

BY FRANCIS L. BAIN*

The demand for the teaching of manual arts in the high school is founded upon the need of the pupil. It has grown out of the dissatisfaction of the people with the narrow, one-sided curriculum of the secondary school. The old measure of culture, training in the classics, has given way to broader lines of development. Today that man is considered cultured who can ably utilize the forces of civilization of his own time, not of a few centuries back.

We have come to realize that the disciplinary value of work which makes for vocational training may be greater than that which comes from the so-called cultural studies. The skilled artisan contributes more to the prosperity of the country and the uplift of his community than the college graduate without the adaption to any line of work. The one goes straight to the mark; the other flounders hopelessly in trying to find himself and his work. The world never needed leaders more than now—leaders of education, of skill, of honesty and of courage, and real leadership is as necessary in the trades as in any department of commercial activity.

The study of the manual arts in the high school has four distinct values:

1. It helps the student to find himself. Nearly two-thirds of the pupils in our public schools leave school at the age of sixteen or earlier. This is due to lack of inclination to study further or the necessity of bread-winning. They usually enter and remain in the ranks of unskilled labor, at the lowest wages and in most unstable positions. These pupils need the stimulus of courses that lead to quick or greater efficiency in wage earning. Many students have no definite aim at entrance to or even at graduation from high school. Chance determines their vocation, and usually chance makes a mistake. Manual arts help the uncertain boy to determine his bent and to choose more intelligently his vocation.

2. It enables the high school to adapt itself to the needs of the community. That high school is best that best meets the needs of its immediate locality. Communities may be divided into commercial, manufacturing and agricultural, and vocational courses may be so shaped as to meet the needs of any or all.

3. It vitalizes the teaching of mathematics and science by showing and using their practical application. Algebra and geometry, physics and chemistry no longer appear simply as studies in a course, but they become a means to an end; as useful tools, they provide studies of living interest.

4. It furnishes direct vocational training. A high school course in manual arts does not aim to make a foreman or a journeyman workman of a boy. Such work must be left to the shop. The school should be able, however, to turn out a student who ranks with advanced apprentices, and whose academic and technical training has been such that, if he has otherwise the right qualities, he can forge rapidly to the front in his chosen trade.

Work in the manual arts, undertaken for vocational training, should not be undertaken without well equipped shops and skilled special teachers. To play with manual arts is to invite contempt for them at the start. It may not be necessary or advisable to attempt many lines of work, but what is undertaken should be done thoroughly and as nearly as possible under shop conditions. Every stroke should count toward definite construction. There is more mental discipline, more technical skill, and far greater interest, in making something of mechanical value, than in the doing of "exercises" or in playing with tools. Work must be the key-note of all the arts.

Technical courses should have all the time possible given to their distinctive work. A few hours per week will not develop skill, vocational or otherwise; all the time that can be given will not develop much; that must come from the experience of the work-shop.

All the attendant academic studies, English, mathematics,

* Director, Department of Technical and Manual Training, Everett High School, Everett, Mass.

science and drawing can be made to bear directly upon this technical work, and so be vitalized, because the pupil appreciates their use and application.

In all the manual arts courses the work should be so arranged as to give each student in junior and senior years opportunity to choose between additional shop and academic work and to specialize along chosen lines. This allows the student to fit himself for higher technical schools, if he so desires, or to gain additional skill in wood-working, forging or machine-shop practice.

Manual arts is not an educational cure-all. It will not revolutionize the work of high schools. Pupils will not all flock to its study. It will be subject to many of the same defects and limitations as other courses and it will have additional limitations of its own; but it will spell opportunity, educationally and vocationally, to a large class of students who have hitherto received too little attention.

* * *

TAP DRILL SIZES

BY F. W. GATES*

In the January number of MACHINERY, Albert A. Dowd gives an interesting formula for determining the size of drill to use before tapping. Boiled down, he advocates 80 per cent depth of thread in the nut for machine tapping and 90 per cent for hand tapping. In this connection, some of the results obtained in the Wells Brothers Co.'s experimental laboratory may, perhaps, prove of interest.

Using U. S. standard threads, an ordinary cold-punched nut even, reamed out until it has but 50 per cent depth of thread, will not strip. Neither will it strip the tops off the bolt thread—*the bolt always breaks*. In fact, we have never found a metal tough enough to strip a nut thread having 50 per cent of full depth. If, in tapping, we produce nut threads which are stronger than the bolt, with a reasonable margin of safety, nothing is gained by attempting to make fuller or deeper nut threads. Such attempts only make unnecessary waste in tapping operations. We have demonstrated that a full depth of thread in a nut is *practically* no stronger than 75 per cent depth of thread. We have also demonstrated that 75 per cent depth of thread is only 20 per cent stronger than 50 per cent depth of thread. (Theoretically it might seem to be 50 per cent stronger.) From 75 per cent depth of thread, the power required to tap rapidly increases as a full depth of thread is approached.

The breaking strain of the average ½-inch, 13 U. S. standard commercial tap as found on the market is approximately 700 inch-pounds. Tapping common cold-punched nuts with *new* taps, the *average* power required is as follows, using mineralized lard oil as a lubricant:

75 per cent depth of thread...... 199 inch-pounds
90 per cent depth of thread...... 365 inch-pounds
100 per cent depth of thread...... 575 inch-pounds

Therefore, in attempting to tap out a full depth thread, the strain is too close to the breaking strain of the tap to be advisable. As the tap wears and becomes dulled, the power required fully equals the breaking strain, and "away goes the tap." There is practically no margin of safety. But in tapping 75 per cent depth of thread there is a factor of safety of practically 3.

On a ½-inch tap the difference in cut between 75 per cent and 90 per cent depth of thread is only 0.0075 inch; yet, under the favorable conditions given, the power required to make the deeper thread is nearly double. Tapping dry, or using machine oil, the ratio of increase in power is very much greater than that given above.

Inasmuch as with 75 per cent depth of thread the nut is stronger than the bolt—so much stronger as to break the bolt without even "starting" the nut threads—and, inasmuch as 75 per cent depth of thread possesses practically all the strength of a full depth thread, one can readily see that the 75 per cent depth is much preferable to the full depth. For instance, grit or dirt, in the case of 75 per cent depth, works down to the root of the bolt threads and gives no trouble. In a full depth thread it is forced into the walls, creating unnecessary friction and destroying the smoothness of the threads.

* Advertising Manager, Wells Brothers Co., Greenfield, Mass.

For these and many other reasons, we have for some time been advocating drill sizes for taps which yield 75 per cent depth of thread in the nut or tapped hole. However, as commercial drills do not exactly meet these figures, by using the next larger commercial size, good results are obtained. The use of these sizes of drills is proving a revelation in tapping economy to many shops.

TABLE OF TAP DRILL SIZES

Nominal Size, U. S. S.	Root Diameter, U. S. S.	Drill Sizes Recommended		
		For 75 Per Cent Thread, Exact	Next Largest Commercial Size	
			In Decimals of an Inch	Commercial Designation
$\frac{1}{4}$	0.1850	0.201	0.201	No. 7
$\frac{5}{16}$	0.2403	0.258	0.261	G
$\frac{3}{8}$	0.2936	0.314	0.316	O
$\frac{7}{16}$	0.3447	0.368	0.368	U
$\frac{1}{2}$	0.4001	0.425	0.4330	11 mm.
$\frac{9}{16}$	0.4542	0.481	0.4843	$\frac{31}{64}$ inch
$\frac{5}{8}$	0.5069	0.536	0.5468	$\frac{35}{64}$ inch
$\frac{11}{16}$	0.5700	0.599	0.6093	$\frac{39}{64}$ inch
$\frac{3}{4}$	0.6201	0.652	0.6562	$\frac{21}{32}$ inch
$\frac{13}{16}$	0.6831	0.715	0.7187	$\frac{23}{32}$ inch
$\frac{7}{8}$	0.7307	0.767	0.7677	$19\frac{1}{2}$ mm.
$\frac{15}{16}$	0.7929	0.829	0.8437	$\frac{27}{32}$ inch
1	0.8376	0.878	0.8858	$22\frac{1}{2}$ mm.
$1\frac{1}{8}$	0.9394	0.986	1.0000	1 inch
$1\frac{1}{4}$	1.0644	1.111	1.1220	$28\frac{1}{2}$ mm.
$1\frac{3}{8}$	1.1585	1.213	1.2187	$1\frac{7}{32}$ inch
$1\frac{1}{2}$	1.2835	1.338	1.3386	34 mm.

Machinery

It is a fact, readily demonstrated, that 90 per cent of commercial taps break before they have been used long enough to require sharpening. This tremendous breakage is seldom the fault of the taps. Strains are often brought to bear upon taps which no metal can possibly withstand. The most important cause is the use of too small holes. It is not by any means uncommon to find taps used in holes which are smaller than the root diameter of the tap.

It would be a good thing if the manufacturers of both taps and drills would get together and publish uniform tables of drill sizes for tapping. At present there is no agreement, and the sizes recommended vary all the way from 75 to 101 per cent depth of thread. For instance, in one very commonly used table the drill size recommended for ½ inch, 13 U. S. standard, yields 94 per cent depth of thread. On 9/16 inch, the next larger tap, a size is recommended which yields 101 per cent depth of thread. "Somebody" started it, and custom continues it without questioning its advisability.

Just by way of confirming the conclusions we had reached through our own tests, we recently took a poll of leading manufacturing concerns throughout the country, from makers of very fine machine work to farm implements, including electrical fixtures, calculating machines, commercial bolts and nuts, harvesting machinery, etc. With one exception, their practice is to use drills which yield 75 per cent depth of thread in the nut, or less; one very prominent shop uses 50 per cent depth of thread in tool steel with good results. We therefore earnestly advocate 75 per cent depth of thread in the nut as a maximum where standard thread systems are used (such as U. S., S. A. E. and A. S. M. E.), because it gives all the strength of a deeper thread and brings about a much needed economy in tapping operations.

* * *

HIGH-GRADE BEARING METAL

A high-class bearing metal is prepared as follows: Melt 7 parts copper at as low a heat as possible, then add 25 parts antimony and 200 parts tin. This mixture is cast in iron ingot molds. It is then re-melted and to each five pounds of the ingots is added eight pounds of tin. This second alloy is cast in bars to suit the requirements.

THE VALVE PROBLEM ON GASOLINE ENGINES*

UNDESIRABLE EFFECTS OF GRINDING THE VALVES AND DESIGNS THAT ELIMINATE THE DIFFICULTY

BY M. TERRY†

What is the proper size of valves for a gasoline engine of a given bore and stroke? What is their proper lift? These, perhaps, are among the most puzzling questions which confront both mechanics and designers, but the vast majority do not even understand the principles underlying the correct solution of the problem. Not knowing what end to begin with, they invariably fall back on "common practice." Just what common practice is and how it works might be illustrated by an incident which occurred a few months ago. A fellow draftsman who was laying out a new gasoline engine came to consult the writer in regard to several details connected with its design; valve lift was one of them.

"And what would you suggest for the valve lift?" he asked.

"Well," I replied, "that is no easy question to answer offhand; it requires a close study of your engine and, perhaps, considerable figuring." Close study and considerable figuring never did appeal to this gentleman; so, entirely ignoring my reply, he continued:

"Jones Motor Works, on their Model X engine, which is 1/4 inch smaller in bore and 1/2 inch longer in stroke than mine, use 1/4 inch lift; and the Smith engine, which is practically identical with mine except that the stroke is 3/4 inch longer, use 5/16 inch lift. I am going to make my lift 9/32 inch; do you think that will be all right?"

"Well, it ought to be," I replied dryly, fully realizing the uselessness of further argument.

In selecting 9/32 inch for his valve lift he apparently showed excellent judgment, for the chief draftsman heartily approved of his method of procedure. "That's fine," he declared, "but how do you know that Smith and Jones are right?" While the question struck the nail squarely on the head, it was asked more with the object of teasing the man than to seriously question his method.

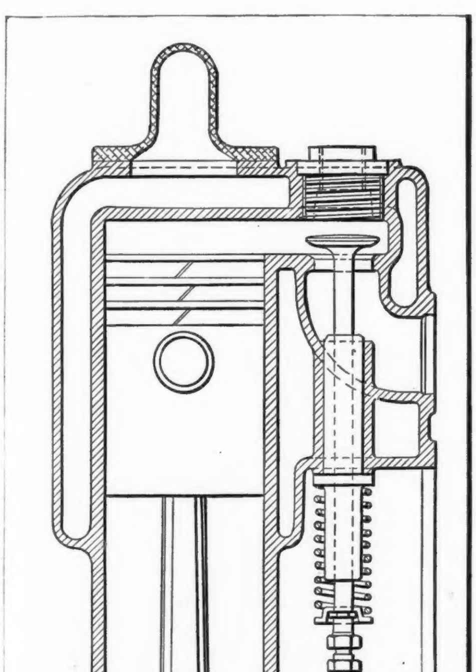


Fig. 1. Arrangement of Valve Gear on L-type Engine

"I do not know whether they are right or wrong," was the reply, "but I do know that their cars sell like hotcakes."

Time and again the writer has had a chance to observe that if So-and-So does this or that, it is a far stronger and more convincing argument than pages of figures. Now our own firm was by no means a "second-rater," and both the Jones Motor Works and the Smith Motor Car Co. were undoubtedly doing the very same thing with our valve lift.

"Our design follows common practice," said the chief engineer to the general manager in an effort to convince the latter that he had in him a man of safe ideas. This was really a mis-statement; the right name for this sort of thing is the "common practice of designing" and not "designing along the lines of common practice."

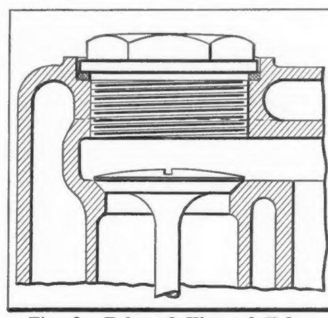


Fig. 2. Enlarged View of Valve Head on its Seat

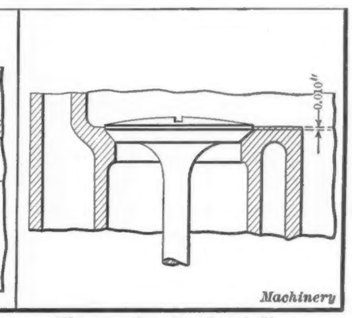


Fig. 3. Result of grinding Valve Seats

Proper Design of Valve Head and Valve Seat

Neglecting for the present the question of valve size and of valve lift, we shall turn our attention to the problem of properly proportioning the valve head and its seat. There is absolutely no theory in connection with this important point; it all hinges on a few practical considerations, and can be easily understood by any man with a mechanical mind. Fig. 1 shows the general arrangement of the valve gear on what is known as the L-type engine, and Fig. 2 shows an enlarged view of the valve head on its seat. The design is wrong. As long as the motor is new, no trouble will be experienced, but as soon as the valve seats pit and wear, the owner of the motor is compelled to grind them, and in doing so the state of things shown in Fig. 3 will result. Assuming that the valves now seat 0.010 inch lower than they did when the motor was new, that the valve lift is 0.250 inch and clearance is 0.005 inch, let us see what happens. The total valve lift, of course, is not changed, as it is governed entirely by the shape of the cam. The effective valve lift, *i. e.*, that part of the total lift during which the gases are free to pass in and out of the cylinder, is now only 0.235 inch—a reduction of 4 per cent. When the motor was new the effective lift was 0.245 inch. This loss of 4 per cent, appreciable as it is, is by no means the end of the story. It will be observed that the exhaust commences later and ends sooner in the stroke; and the incoming fresh charge is similarly affected. The extent of this change of original timing, which is seldom appreciated, will be presently shown.

Let the original timing be as shown in Fig. 4. The base circle of the cam is 1.000 inch in diameter; the roller diameter is 1.000 inch; and the clearance between the valve stem and the valve lifter is 0.005 inch. The writer is assuming a certain amount of knowledge on the part of his readers in regard to the construction and operation of gas engine valve gears. Those who are unfamiliar with it are referred to another article entitled "Dynamics of Gas Engine Cams" published in the engineering edition of MACHINERY for November and December, 1912, where the problem was thoroughly discussed and the relation between the clearance and the clearance angle was fully explained. In Fig. 5, the cam and its roller are shown in position where the latter is just on the point of rising. Therefore:

$$oc = 1.000 \text{ inch.}$$

At c_1 the backlash is closed and the valve is just on the point of rising. Therefore:

$$oc_1 = oc + \text{backlash} = 1.000 + 0.005 = 1.005 \text{ inch.}$$

c_2 is the point where the effective lift of the valve commences, *i. e.*, where the gases are first commencing to leave

* For additional information on the subject of gasoline engine valve gears and allied subjects, see "Dynamics of Gas Engine Cams," published in MACHINERY, November and December, 1912, and other articles there referred to.

† Address: 1063 Manning St., Flint, Mich.

or enter the cylinder, as the case may be. According to our assumption:

$$oc_2 = oc_1 + 0.010 = 1.005 + 0.010 = 1.015 \text{ inch.}$$

$$\text{Now } \cos \alpha = \frac{1.000}{1.005} = 0.995$$

$$\alpha = 5 \text{ degrees } 45 \text{ minutes}$$

$$\cos \beta = \frac{1.000}{1.015} = 0.985$$

$$\beta = 9 \text{ degrees } 45 \text{ minutes}$$

$$\beta - \alpha = 9 \text{ degrees } 45 \text{ minutes} - 5 \text{ degrees } 45 \text{ minutes} = 4 \text{ degrees.}$$

Thus the inlet and exhaust valves start 4 degrees later and close 4 degrees earlier than they did when the engine was new. This loss, of course, is doubled on the crank circle, since the crankshaft of a four-cycle engine revolves twice as fast as its camshaft. The loss, then, amounts to 8 degrees at opening and 8 degrees at closing of the valves, and the modified timing diagram is as shown in Fig. 6. It will be observed

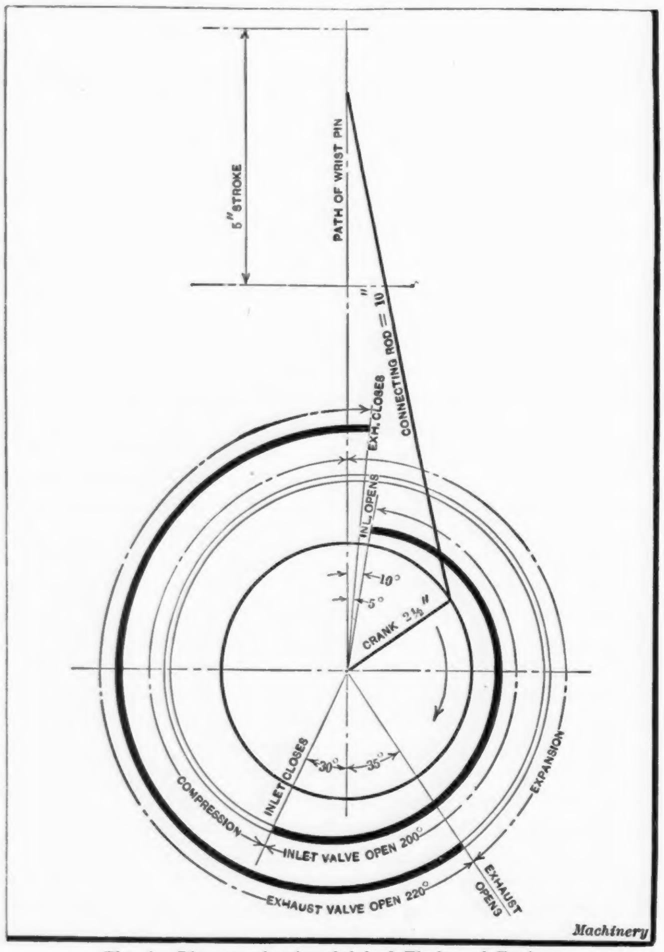


Fig. 4. Diagram showing Original Timing of Engine

that the loss is appreciable, amounting, as it does to $\frac{16 \text{ degrees}}{200 \text{ degrees}}$ or 8 per cent for the inlet valves, and $\frac{16 \text{ degrees}}{220 \text{ degrees}}$ or 7.27 per cent for the exhaust valves. The combined loss due to the shorter effective valve lift and shortened duration of the period of valve opening is shown very strikingly in Fig. 8, which is the time-lift curve of the inlet valve. The area bounded by the curve and the clearance line can be taken to represent the charge admitted to the cylinder when the engine was new, as explained in the article previously referred to. Similarly the area bounded by the curve and the effective lift line represents the charge admitted after the wear of 0.010 inch has taken place. The shaded area represents the volumetric loss which amounts to about 7.65 per cent for the exhaust and 8.5 per cent for the inlet valve; these results were obtained by means of a planimeter. The volumetric loss, as the term is ordinarily understood, occurs only on the inlet or suction stroke; a similar loss, however, which may be termed loss in scavenging action, takes place on the exhaust stroke. Either loss is undesirable, as it results in a falling off of the power developed by the engine.

The preceding discussion will probably account in many cases for the oft-heard complaint that the horsepower of certain automobile engines seems to drop off with their age, in spite of replacing the old, worn piston rings by new ones, and thereby restoring the compression pressure to its original value. The valve seat being comparatively inaccessible, an average repair man is not likely to notice the little ridge formed around it, and of those who would notice it not one in a hundred would realize its significance and attempt to remedy the trouble. As a matter of fact, the trouble in most cases is not very easily remedied, and the reason for it is not far to seek. The plug above the valve (Figs. 1 and 2) is only a plug; it is simply put there to close the hole formed in order to permit machining the valve seat and assembling the valve. Since the performance of any gasoline engine depends to a great extent upon the size of its valves, the latter are invariably made as large as possible, and in the construction shown in Fig. 2 the valve head is made only a few thousandths inch smaller than the threaded hole above it. Evidently the tool for chamfering the valve seat cannot be made any larger

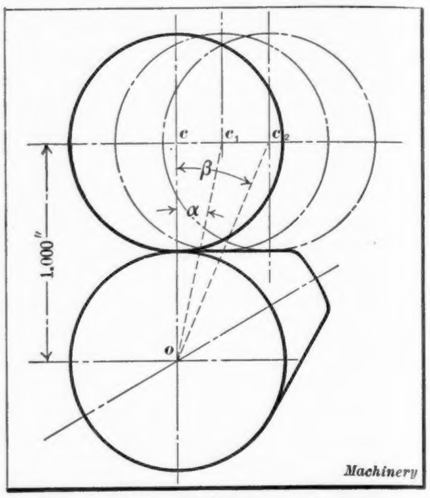


Fig. 5. Valve Lifting Cam with Roller ready to rise

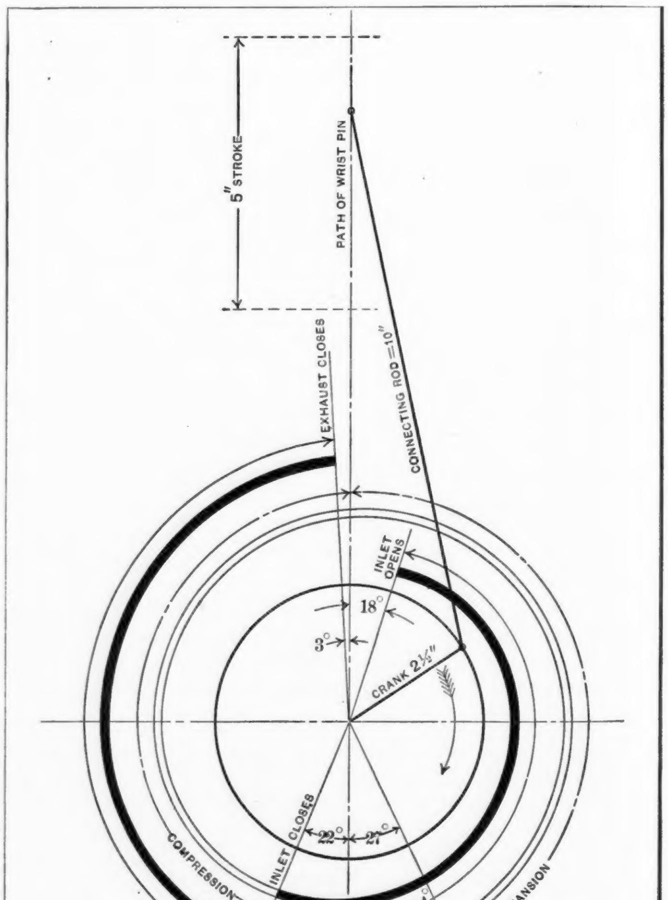


Fig. 6. Modified Timing Diagram showing Result of grinding Valve Seats

than the valve head, and hence the difficulty which must inevitably face the repair man, should he take it into his head to remove the ridge. There is only one way—so far as the writer knows—of remedying the trouble, and that is by means of a fly-cutter. Care must be taken, however, to center the shank of the cutter properly by means of some special fixture; otherwise the valve seat and the valve stem guide hole will not be concentric, and gas leakage and poor compression are sure to follow. It will be seen from the foregoing that small and insignificant as the question of proper proportioning of the valve head and its seat at first appeared to be, it is far-reaching in its results, and surely deserves some consideration on the part of designers. The trouble we have just been investigating is very easily remedied—if it is recognized—on engines equipped with removable valve cages as shown in Fig. 7, which is the Buick design, and on the L-type engines having a detachable cylinder head like the Ford.

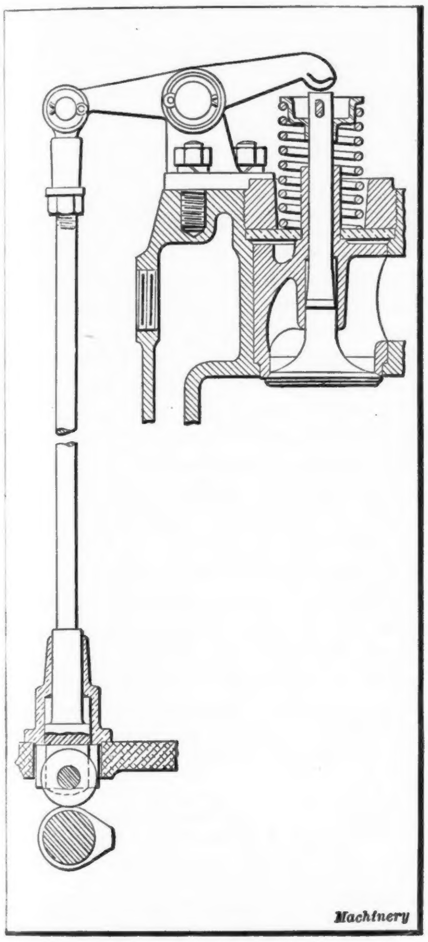


Fig. 7. Removable Valve Cage used on Buick Engine

A correct design is shown in Fig. 9, where the outside diameter of the valve D is made from 1/16 to 3/32 inch larger than D_1—the outside of the bevel seat. The width of the seat W generally varies from 3/32 to 3/16 inch. The diameter at the lower edge of the valve head is made equal to the bore of the throat; occasionally it is from 1/64 to 1/32 inch smaller. This is all right; but under no circumstances should it be made larger than the bore of the throat, for a flat seat will eventually be formed at the inner edge of the bevel seat, thus resulting in a tortuous path for the gases—a condition admittedly undesirable on any gas engine. In addition to this, the contact between the valve head and its seat will take place in two planes with the probability of being far less perfect than it was before.

The Bore of the Cylinder vs. the Bore of the Throat

Any mechanically inclined person surely recognizes the fact that there must be a certain direct relation between the size of any given engine and the size of its valves; the larger the cylinder, the larger the valves. But what is the exact relation between the two? As stated before, the valves are invariably made as large as a given design will permit. This, however, does not settle the problem; on the contrary, the important test of any given design is the size of its valves, and should calculations show the latter to be too small, the design must be changed to permit the use of larger valves. In the absence of extensive experimental data on the subject or even of a simple and plausible theory which would take into account several important factors, the designers of today reduce the whole question of proper size of valves to a simple numerical relation between the bore of the cylinder and the bore of the throat. Let:

D = diameter of cylinder bore in inches;
S_p = average piston speed in feet per minute;
L = length of piston stroke in inches;
t = duration of piston stroke in minutes;
V = piston displacement in cubic inches;
d = diameter of bore of throat in inches;
S_g = average velocity of gases at the throat in feet per minute;

$$V = \frac{\pi D^2}{4} \times L = \frac{\pi D^2}{4} \times 12 S_p \times t \qquad (1)$$

Since the same volume V has passed through the valve throat and neglecting the reduction of the passage area due to the presence of valve stem:

$$V = \frac{\pi d^2}{4} \times 12 S_g \times t \qquad (2)$$

Therefore

$$\frac{\pi d^2}{4} \times 12 S_g \times t = \frac{\pi D^2}{4} \times 12 S_p \times t \qquad (3)$$

$$D^2 S_p t = d^2 S_g t \qquad (4)$$

$$\frac{D^2}{d^2} = \frac{S_g}{S_p} \qquad (5)$$

It is common practice among designers to so proportion D and d that S_g shall not exceed the value of 6000 feet per minute at the time the average piston speed S_p is 1000 feet per minute.

Formula (5) then reduces to the form:

$$\frac{D^2}{d^2} = \frac{S_g}{S_p} = \frac{6000}{1000} = 6$$

$$6\,d^2 = D^2$$

$$d = \frac{D}{\sqrt{6}} = \frac{D}{2.45} \qquad (6)$$

For example, if $D = 3$ inches, $d = \dfrac{3}{2.45} = 1.225$ inch.

The bore of the throat should not be less than 1¼ inch diameter and this determines the size of the valves. Almost any multiple-cylinder engine may be redesigned to accommodate larger valves, but if this advantage is secured at the expense of spacing the cylinders farther apart, thereby increasing the overall length of the engine and making a corresponding increase in its weight, as well, the procedure is not worth while. This is especially true in the case of automobile engines, where weight is a question of prime importance. It must not be supposed, however, that if an engine with a 3-inch cylinder bore and 1¼-inch valves has its piston moving

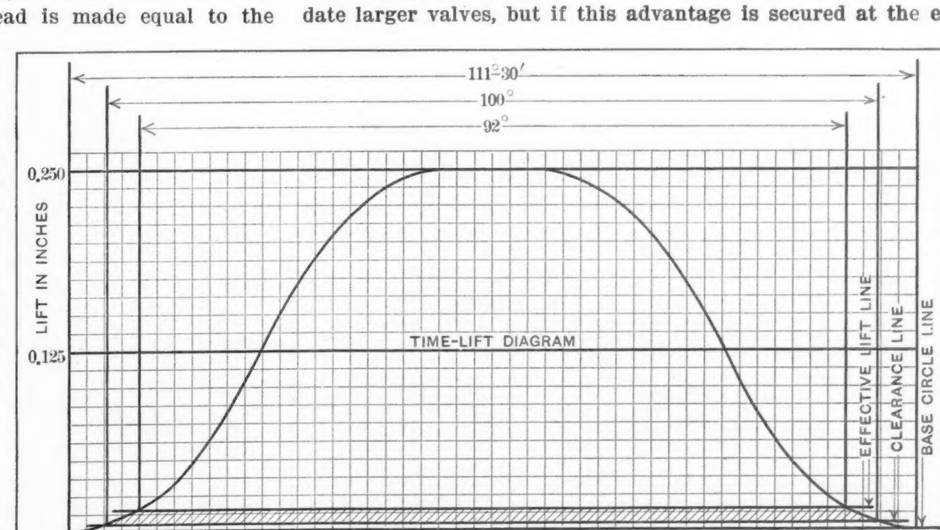


Fig. 8. Time-lift Diagram showing Combined Loss due to Shorter Effective Valve Lift and Shorter Period of Valve Opening

at the rate of 1000 feet per minute the average velocity of gases is actually 6000 feet per minute. It is very unlikely that it is anywhere near this figure. In the first place, Equation (4) is not strictly true, because the suction stroke of the piston lasts through 180 degrees, while the inlet valve, according to Fig. 4 in our case, is open through 200 degrees of the crank circle. In the second place, the statement that the volume V, which is equal to $\frac{\pi D^2}{4} \times L =$ piston displacement, passes through the valve throat, is far from being true. The weight of the gases drawn into the cylinder is equal to the weight of the gases passing through the valve throat, but their volumes are not equal. This inequality in volume arises from two causes: first, the temperature of gases at the instant they fill the cylinder must be considerably higher than at the time of their passage through the throat; second, from the analogy between aero- and hydro-dynamics, we know that, neglecting frictional losses, the total energy of a given weight of moving gas in the cylinder is equal to the total energy of the same weight of gas at the throat. The total energy is the algebraic sum of the velocity and pressure energies; and as the velocities of the gases in the cylinder and at the throat are different, it is clear that the gas pressures cannot be the same. As the gases are in a different state as regards their pressure and temperature, their volumes are more than likely to be different.

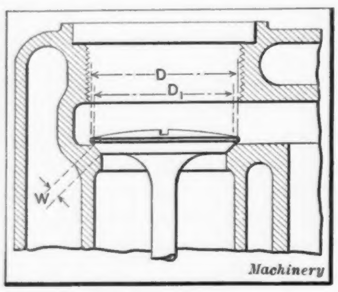


Fig. 9. Design of Valve with Seat that can be properly ground

There is yet another cause which tends to make the actual velocity of gases different from that allowed in the derivation of Formula (6), but its discussion will be postponed for the present. The natural conclusion is that Formula (6) is based on assumptions which are far from actual working conditions. Nevertheless, on account of being simple, it serves the purposes of design as well as any empirical formula. With the accumulation of experimental data in the years to come, this formula will undoubtedly be considerably revised and extended. As an empirical formula, $d = \frac{D}{2.45}$ falls short of meeting the requirements of a thoughtful designer. Let us consider, for example, a 3 by 4 and a 3 by 6 engine; according to our formula the throat opening on both engines should be 1¼ inch in diameter. Now, let the pistons of both engines be moving at precisely the same speed, say 1000 feet per minute. Since the stroke of the first engine is 4 inches, its piston will cover 1000 feet in 1500 revolutions of the crankshaft. The piston of the second engine will cover the same distance in 1000 revolutions of its crankshaft. Since the Otto cycle is completed in two revolutions of the crankshaft, there will be 750 cycles per minute on the 3 by 4 and only 500 on the 3 by 6 motor. The column of fresh gases must be started and stopped 750 and 500 times per minute, respectively, on the two engines; the same applies to exhaust gases. Also, the duration of valve opening, and, hence, the duration of gas flow is 50 per cent greater on the 3 by 6 than it is on the 3 by 4 motor. The result is that at the same piston speed, the former engine is bound to excel the latter. This is one of the causes of the recent remarkable performances of certain long-stroke racing motors, both here and abroad, and of the increasing popularity of the long-stroke engine for pleasure cars. Our formula $d = \frac{D}{2.45}$, failing as it does to take into account the stroke of the engine, cannot give entire satisfaction.

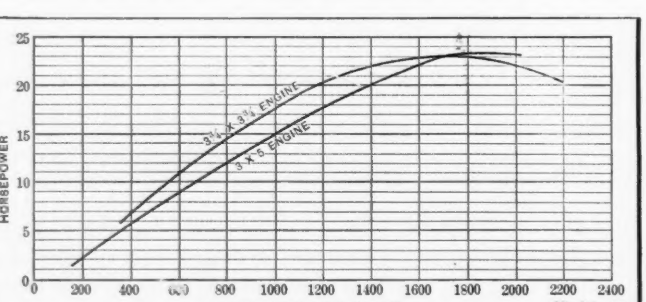


Fig. 10. Power Curves of Engines subjected to a Commercial Test

Again, the constant 2.45 in the preceding formula and the gas velocity of 6000 feet per minute on which this formula is based, are not applicable to all engines. Certain types of engines can accommodate valves that will easily permit as low a gas velocity as 5000 and even 4000 feet per minute, while others necessitate a velocity closely approaching 10,000 feet per minute. Yet, in spite of this great disadvantage, an engine with valves in the head having 10,000 feet per minute for its gas velocity, may show a better economy than an L- or T-head engine with 5000 feet per minute for its gas velocity. This phenomenon is generally accounted for by the fact that the combustion chamber of the valves-in-the-head engine has less radiation area per unit volume than that of the L- or T-head engine, and consequently more heat units are converted into useful work and fewer lost by radiation. Whether this generally accepted explanation of the phenomenon is correct or not, the fact remains, and hence, the apparent necessity of having a distinct constant or permissible rate of gas flow, for each type of engine. No better proof of this necessity of differentiating between engines of distinct types can be given than by making a comparison of an L-head and a valves-in-the-head engine, and studying their power curves as obtained from an actual test. The curves as shown in Fig. 10 are the result of an ordinary commercial and not a scientific test; they also represent the brake and

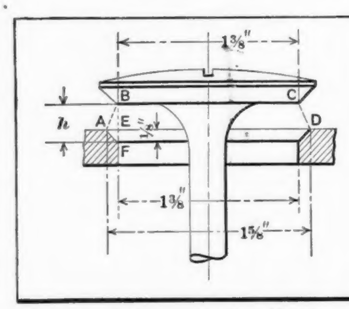


Fig. 11. Diagram showing Method of determining Valve Lift

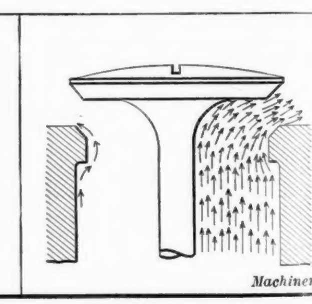


Fig. 12. Diagram showing Effect of "Vena Contracta"

not the indicated horsepower. Strictly speaking, the latter is preferable for our purpose. Also, the two engines happen to have somewhat different piston displacements, this circumstance slightly favoring the larger engine for no other reason than that large engines invariably show better economy or generate more power, displacement for displacement, than the smaller ones. These minor shortcomings, however, detract but little from the value of our comparison. The principal dimensions of the two engines are tabulated in Table I. From a study of this table, particularly items 8, 9 and 14, one would be led to expect greater output, displacement for displacement, from the 3 by 5 rather than from the 3¾ by 3¾ engine. Such, however, is not the case, as can be seen from their power curves (Fig. 10) and Table II which is derived from them.

A more recent test of the two engines conducted with the same size and make of carbureter has brought out a still better performance, displacement for displacement, of the 3¾ by 3¾ engine, which maintained the lead on the 3 by 5 engine up to 1800 revolutions per minute, at which speed their performance curves crossed. By making a comparison on the basis of the same rotary speed we eliminate the factor of cycles and duration of valve opening, thus leaving only two main factors unbalanced, namely: rate of gas flow and horsepower per unit displacement. In engines of the same type, lower rates of gas flow are invariably accompanied by greater horsepower per unit displacement. Referring to Table II and the statement made in regard to it, it will be observed that at low, moderate and even high speed, the

valves-in-the-head engine shows an equal and even better performance than the L-head engine, in spite of its almost 50 per cent greater rate of gas flow. This proves it to be a fallacy to prescribe the same rate of flow, *i. e.*, 6000 feet per minute at the piston speed of 1000 feet per minute for engines of all types.

Valve Lift

Last, but not least, comes the problem of valve lift. Assuming that our valve throat is properly proportioned, it remains—as a matter of common sense—not to lose the advantage thus gained by restricting the passage and thereby choking the gases anywhere along the line—something that can be very easily accomplished by providing an insufficient valve lift. The problem consists of so proportioning the lift

TABLE I. DIMENSIONS OF L-HEAD AND VALVES-IN-THE-HEAD TYPES OF ENGINES SUBJECTED TO TEST

	Valves-in-the-head	L-head
1. Type of engine	Valves-in-the-head	L-head
2. Number of cylinders	4	4
3. Bore = D inches	$3\frac{3}{4}$	3
4. Stroke = L inches	$3\frac{3}{4}$	5
5. Area of cylinder bore = $\frac{\pi D^2}{4}$ square inches	11.045	7.069
6. Total piston displacement = $4 \times \frac{\pi D^2}{4} \times L$ cubic inches	166	141
7. Revolutions per minute corresponding to piston speed of 1000 feet per min.	1600	1200
8. Number of cycles per minute at piston speed of 1000 feet per minute	800	600
9. Duration of valve opening in per cent assuming same timing	100	133
10. Bore of valve throat in inches	$1\frac{1}{4}$	$1\frac{5}{8}$
11. Diameter of valve stem in inches	$\frac{3}{8}$	$\frac{3}{8}$
12. Net area of valve throat in square inches	1.117	1.375
13. Area of cylinder bore ÷ net area of valve throat	9.9	5.14
14. Rate of gas flow in feet at a piston speed of 1000 feet per minute	9900	5140

Machinery

(h in Fig. 11) that the area of the frustum of the cone of revolution $ABCD$ be made equal to the net area of the throat.

Net area of throat $= \frac{\pi}{4}(1.375^2 - 0.375^2) = 1.375$ square inch.

Area of the frustum $ABCD = AB \times \pi \times \frac{1\frac{3}{8} + 1\frac{5}{8}}{2} = 4.71\ AB$.

$$AB = \frac{1.375}{4.71} = 0.292 \text{ inch.}$$

$$BE = \sqrt{AB^2 - AE^2} = \sqrt{0.292^2 - 0.125^2} = 0.264 \text{ inch.}$$

Valve lift $= h = BE + EF = 0.389$ inch.

The valve shown in Fig. 11 was used on the 3 by 5 engine previously described. The lift as actually employed was only 7/32 inch or about 56 per cent of the calculated amount. It may seem absurd, but this dwarfing of the lift is being done on a wholesale scale—in most cases unconsciously—by the followers of the "common practice" theory, who have no sounder arguments to advance than that "they all do it." The true reason for the dwarfing of the valve lift, whenever it is done knowingly, of course, and not in the act of merely copying someone else, lies in the fact that high lifts are invariably associated with increased stresses on the parts comprising the valve gear and their rapid wear. This consideration is of so serious a nature that it cannot be ignored. But apart from this purely mechanical reason there is yet another reason, seldom recognized, but which nevertheless exists. The writer is referring to the presence of what is known in hydraulics as *vena contracta* in the throat passage (Fig. 12) which materially reduces the net area of the throat. Thus, in the case under discussion, the theoretical lift of 0.389 inch is excessive, since our theory did not take cognizance of the existence of vena contracta. But there is also a possibility of increasing the lift from 7/32 to 9/32 or even 5/16 inch and securing better output from the engine without incurring excessive wear of the valve gear. The actual amount of vena contracta, as well as the maximum useful valve lift, can be determined only by experimental means. The existence of vena contracta is the third reason alluded to in the early part of this article when it was stated that the actual gas velocity at the throat differs from the calculated one.

There is one more thing in connection with valve lifts which the writer feels is worthy of mention. Let us take, for example, an engine having 1½-inch valves, with a lift of ⅜ inch. Now, suppose we desire to redesign this engine and increase its valves to 2 inches. The increase of area at the throat is proportional to the square of its diameter; and, in order to maintain the same advantage at the valve seat, the valve lift must be increased roughly to ½ inch. If the latter expedient be considered undesirable, the lift need not be changed, but the net gain, then, will be proportional only to the increase of the throat diameter. Now, the same engine can be redesigned along somewhat different lines. Suppose that quieter running and reduction of wear on the valve gear is the end aimed at, the power performance being entirely satisfactory. This can be secured, as before, by increasing the valve diameter from 1½ inch to 2 inches, and reducing the lift from ⅜ to 9/32 inch, thus leaving the passage area at the valve seat practically unaltered.

Judging by the ratio of the cylinder bore to the valve throat alone, as outlined in the earlier part of this article, the performance of the two engines should be identical—something that is very unlikely to happen owing to the great difference in passage areas at their valve seats. It will thus be seen that the valve throat alone does not form a good criterion of the probable engine performance. Indeed, the practice of comparing and designing engines on the basis of the ratio of the two bores alone may be considered as misleading and unfortunate. It should rather be based on the passage area at the valve seat, the exact amount of which depends both on the diameter of valve throat and the valve lift, the latter being so limited by practical considerations that there is absolutely no danger of its being made too large at the expense of the throat. When the usual method of computation is followed, the valve throat might be, and generally is, made too large at the expense of the lift.

In conclusion, the writer wishes to say that, while the formulas cited are somewhat unsatisfactory, they serve as a guide for one's analysis of gasoline motors. Like all inefficient tools and methods, they cannot be discarded until better ones are found to take their places. The use of formulas alone is practically worthless, unless it is accompanied by the two most important elements in all branches of design: a painstaking analysis and a trained judgment.

* * *

The post office department estimates that the gross receipts from the parcel post business during 1914 will amount to $80,000,000, of which $30,000,000 will be profit. This is a conservative estimate based on the results of the past year. The railroads claim that out of the profits $15,000,000 ought to be paid them in addition to the payments now made for carrying the parcel post mail, but even if that were done the profit realized is a fair one, and not at all in agreement with the opinions of leading business men engaged in the express service who claimed that the government could not carry on the parcel post except at a loss.

TABLE II. COMPARATIVE TESTS OF GASOLINE ENGINES

	800 R. P. M.		1200 R. P. M.		1600 R. P. M.	
Engine	3 by 5	$3\frac{3}{4}$ by $3\frac{3}{4}$	3 by 5	$3\frac{3}{4}$ by $3\frac{3}{4}$	3 by 5	$3\frac{3}{4}$ by $3\frac{3}{4}$
Piston speed in feet per minute	666	500	1000	750	1333	1000
Gas velocity in feet per minute	3425	4950	5140	7420	6850	9900
H.P. per cubic inch of displacement	0.085	0.088	0.126	0.126	0.156	0.136

Machinery

THE TESTING OF MATERIALS*

The ever-increasing intensity of the stresses which the materials of engineering, and especially of metals used in machine construction and structural designs, are expected to withstand, or the energy they are required to transmit, has during recent years necessitated a more and more intimate knowledge of the properties of the materials employed. At the present time, also, a great deal of study is given to the laws which determine the distribution of the stresses in the elements of the mechanical details or structures themselves. It is, however, impossible to deny that the architects of the past, who conceived and carried out such designs as are found in Greece and Italy and in the great Gothic cathedrals, possessed a considerable knowledge of the materials they employed and a keen sense of the stresses to which they submitted them; so did also the engineers and builders who launched upon the ocean the war-ships and merchantmen of the eighteenth century. They did not, however, for the most part, possess a knowledge of accurate calculation, nor of the definite laws necessary for the determination of the elastic properties of their materials. They took for guidance structures of a similar nature that had already proved successful, and when it was a question of undertaking entirely new work, either as regards size or shape, the designer was guided by the sense of fitness which practice is so capable of conferring, although it may sometimes be a mistaken one. Thus the designer was frequently led to over-emphasize the strength, or, if this proved inadequate, to employ means of strengthening his structures by making such means contribute to their ornamentation, as, for example, in the flying buttresses of the Gothic architecture.

The First Definite Investigations into the Strength of Materials

It was not until the seventeenth century, in the era of Galileo, that we find the first attempt made by this great scientist to solve the problem of the resistance of materials. Shortly afterward, Hooke enunciated the fundamental law of stability, which has ever since found application. In the eighteenth century renowned scientists and skillful engineers, such as Bernouilli, Euler, Duhamel, Gauthey, Rondelet, Young, Coulomb and Lagrange, ultimately developed the theoretical investigations which fostered the progress of the mathematical analysis and rational mechanics of the elastic properties of materials. The movement once initiated was never allowed to halt. It was, however, especially toward the improvement of the mathematical theory that the early investigators directed their efforts, but experiments were not lacking during this period. Among those who based their researches on experimental results may be mentioned Morin, Hodgkinson, Stephenson, Fairbairn and Napier.

The early development of the railways gave considerable impulse to the investigations into the strength of materials. The steam railways involved bridges of great length, capable of supporting rolling loads at high speeds. Work of this kind had nothing similar in any previous experience, and new methods and new elements of computation were required; besides the question of cost and maintenance came to the front as never before. Although a great deal was done during this period, it may be said that it was mainly during the course of the last thirty years that the activity in experimental work has become accentuated, owing, on the one hand, to the extraordinary increase in the dimensions of the structures themselves and the loads they are expected to carry, and on the other, to the introduction of new methods of testing and of refinements hitherto unknown. In this connection may be mentioned not only the erection of immense bridges, but of roofs of great span, steamships of immense size, the introduction of reinforced concrete, and the development of the automobile and of aerial navigation. It has therefore been necessary to improve experimental methods, to create more powerful and more accurate testing machines, and to study properties to which no regard was previously paid, such as, for instance, the resistance to shock and to rapidly repeated stresses. It has been necessary to look more deeply into the actual study of the structure of materials and to investigate the various forms of treatment to which metals are subjected before being put into use. A new science, metallography, has arisen, which, although of recent birth, has already been the object of a great deal of research which has endowed the metal industries with a fund of valuable knowledge.

Testing Methods

The most conclusive test, from the point of view of the strength of a structure, is obviously that which consists in subjecting the structure to the maximum stresses it is required to undergo, or even to still greater stresses, so as to insure complete security against all possible accidents. This has been done, for example, in the case of ropes, in boilers, and in bridges. Such tests are of extreme interest from the point of view of the confirmation of the accuracy of the formulas employed in the calculations of the dimensions and of the assumptions that it is often necessary to introduce into these calculations; these tests may often be made to yield valuable information for the completing of similar structures. An attempt has been made to introduce such tests into general practice, and methods and apparatus for same have been developed by Mr. Rabut and applied in France; Mr. James Howard has also developed the methods and apparatus adopted by the Bureau of Standards in Washington. These methods permit of measuring with considerable accuracy the deformation in a structural part; or of the plotting of a diagram showing the deflection in a bridge under the influence of moving loads; or of measuring the angular deviation in structures submitted to wind pressure, or in parts submitted to torsion.

Other investigations have made it possible, in many cases, to substitute hardness tests for tensile tests in iron and steel. Such tests, which may be termed "indirect," often justify their adoption by reason of the simplicity of the appliances required, or their small initial cost, and may often play an important part in practice, even if the information obtained is less accurate than that obtained by direct tests. The latter, however, are the most important and are always adopted when it is necessary to ascertain with the greatest accuracy the elastic properties of a metal, apart from the mode in which it is to be employed. These properties are, in particular, the elastic limit, the ultimate breaking stress, the coefficient of elasticity, the elongation, and the percentage of reduction of area at the breaking point. These tests are carried out on test pieces, that is, on samples cut from the materials to be investigated and specially prepared so as to display the properties under investigation. Experience has shown that this very preparation, as well as the methods of carrying out the tests, considerably influences the results obtained. It has therefore been necessary to fix international standards for uniform methods of testing for each class of material, and in particular for iron and steel. This work of standardization has been carried out by the International Association for Testing Materials, which was formed in 1895. The original aim of this association was to introduce uniform methods for the testing of materials, but while this has only been partly accomplished, the meetings of the association have become occasions of learning new facts and of investigating new methods for testing, rather than classifying those already known throughout the engineering world. When these new methods have been thoroughly established, it will be possible to establish standards which will rest on a scientific basis and yield more exact indications as to the nature of the materials than has been possible in the past.

The Work of the International Association for Testing Materials

As the work of the International Association for Testing Materials is, in a measure, a record of what has been done in recent years throughout the engineering field along these lines, a brief review of the work of the association will afford a comprehensive idea of the present status of the testing of materials. In 1906, at the meeting at Brussels, the association adopted and published a set of methods for the testing

* Abstract of a paper read by Prof. H. Hubert of Liege, Belgium, before the Iron and Steel Institute at Brussels.

of iron, steel, cast iron, copper, metal alloys, hydraulic cement, wood and clay, stoneware and cement pipes. The degree of accuracy in measuring permanent elongation should be 0.001 per cent. The "apparent elastic limit" has been reached when there is a permanent elongation varying between 0.2 and 0.5 per cent and the "proportional elastic limit" has been exceeded when the elongation on increasing the load by 100 kilograms per square centimeter (1422 pounds per square inch) differs by more than 0.0005 per cent from the proportional elongation. The association has also fixed rules for the determination of shock-fractures and resilience.

Metals are subjected to three different classes of tests: chemical, physical and mechanical. The chemical tests are more especially carried out during the process of manufacture in order to ascertain the composition and purity of the products. They are also carried out by the purchaser as a check on the specifications.

Physical tests consist of the examination of the exterior of the products and noting their fracture. The superficial examination may yield a general idea to the expert, but it is the examination of the fracture, especially when the surface can be projected under strong illumination on a screen suitably enlarged, that yields definite indications as to the quality of the steel and cast iron, and the defects that may be present. Of late, this method has been improved by polishing the surface of the fracture and etching it with a dilute solution of acid or preferably with a ten-per-cent ammoniacal copper solution, which allows of a micrographic examination being made immediately. This mode of testing often affords the metallurgist and engineer means for ascertaining the cause of discrepancies manifested in mechanical tests. Micrographical research is a branch of the science of testing in which, more than any other, progress has been made in the past few years, and it has become a trustworthy guide in the manufacture if iron, steel and metal alloys. It has been introduced particularly in works engaged in the construction of automobiles.

The variations in the properties of metals, under the influence of changes of temperature, hardening, tempering and annealing, and the study of magnetic and electric properties, all form branches of the physical testing and have given rise to many investigations.

Mechanical Tests

Mechanical tests were, for a long time, the only ones to which recourse was to be had in the study of metals. They still remain among the most important. These are classified into resistance tests on subjection to tensile, compressive, shearing, bending and torsional stresses, either applied gradually or suddenly, and workshop tests, consisting of bending, hammering down, upsetting and punching. The necessity of using, for the resistance tests, accurate and very costly machines, and of subjecting the test piece to the most careful and therefore the most expensive, preliminary treatment, and entrusting the experiments themselves to an expert staff, renders the time and expense for such tests relatively large, and many investigators have sought to supersede them by more rapid and cheaper treatments, even at the expense of accuracy. Among such methods may be mentioned the Frémont punch-test and the various hardness tests. The Brinell hardness test gives a fairly definite relation between the breaking stress and the hardness number and has the advantage of being capable of application to finished parts without injuring them. Many investigators claim that the relationship between hardness, as measured by the Brinell method, and tensile strength is so close that a coefficient may be determined by which the hardness numeral may be multiplied to obtain the tensile strength. Other observers claim that the results sometimes are very erroneous, and the Brussels congress, in 1906, would not consent to the substitution of the ball hardness test for the tensile test in specifications, but it recognized the value of the process by recommending its use in testing supplies. Later investigators, however, have credited the method with close accuracy. At the meeting of the International Testing Association in New York, 1912, the subject was again discussed, but no definite conclusions were arrived at.

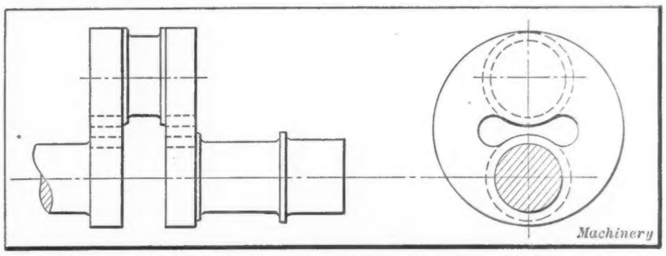
An interesting subject of investigation, relating to the testing of materials, is that of "fatigue" in metals, that is, the giving away of structural parts after they have been in use for a long time under a stress not exceeding that to which they have been continuously exposed. The earliest investigations of this subject were those made by Wöhler between 1857 and 1870. Many methods have been developed for carrying out these tests. It is claimed that 80 per cent of the fractures that actually occur in structural designs are due not to over-stresses, but to fatigue, and much importance, therefore, should be paid to methods of testing metals for continued stresses. The ability of a metal to resist fatigue may be denoted by a "factor of quality" which would indicate not its absolute strength, but the relative number of times that the material could be subjected to a given stress before it would actually break under that stress.

The brief review in the foregoing gives a general outline of the present status of the science of testing materials, for such it might well be termed. The great importance of the work of the International Association for Testing Materials to the engineering field is evidenced by the fact that the association now has about 2700 members from some twenty-five different nations, and that ten governments officially give financial aid to the work of the association. In the United States alone there are nearly 500 members and about as many in Germany. These two countries apparently lead in the interest taken in this important subject.

* * *

IMPROVEMENT IN CRANK AXLES

The illustration shows an improvement which has been applied to crank axles having circular webs, by M. C. Frémont, a French engineer. The stresses in these circular webs are most severe at the center, where the ingot from which the axle is forged is the weakest, and therefore cracks often start at this point, continuing to the periphery. In order to obviate this difficulty, the metal in the center is cut away in the form of the figure 8, thus preventing the starting of cracks. Axles made according to the Frémont



The Fremont Crank Axle with Portion at Center of Web removed

system have been in use in France for four or five years and have given entire satisfaction. In fact, crank axles which showed cracks in the center and were condemned have had the faulty portion cut away and then been put back into service, and have given no further trouble. Axles with circular crank webs have been tested to destruction and have been found to resist greater stress when the webs are cut away in the center, as described, than when the webs are solid.

* * *

An analysis by the *Railway Age Gazette* of the statistics of locomotives ordered during 1913 indicates that the superheater is being applied to nearly all locomotives, except those of the smaller size, and that engines of very large size are much in demand. Specifically stated, 79 per cent of all new steam locomotives of moderate and large size are equipped with superheaters, while in 1912 the percentage was 63. Nearly 63 per cent of the total number of locomotives ordered during 1913 were of the consolidation, Mikado or Pacific type, the consolidation being in the lead of all other types, forming 23 per cent of the total. The Pennsylvania R. R. leads in the use of these engines, 435 having been built for this railroad alone.

THE DIESEL ENGINE*

PRINCIPLE OF OPERATION—CHARACTERISTICS—EFFICIENCY—FUEL

The development of the internal combustion engine has made tremendous strides in the last few years. This is especially true of one type in particular, the Diesel engine. The unparalleled progress of this prime mover, due primarily to the expiration of the basic Diesel patents, has created a wave of enthusiasm. To the careful observer, however, signs of an ebbing of this wave are apparent, and more sober views are rapidly gaining ground. The application of the engine to marine work, especially to vessels of certain classes, has been especially noticeable, and the marine interests of the country have been watching closely the work of the Diesel engines. Other applications of the engine are common, and the Diesel engine of today is likely to be found in almost any line of work.

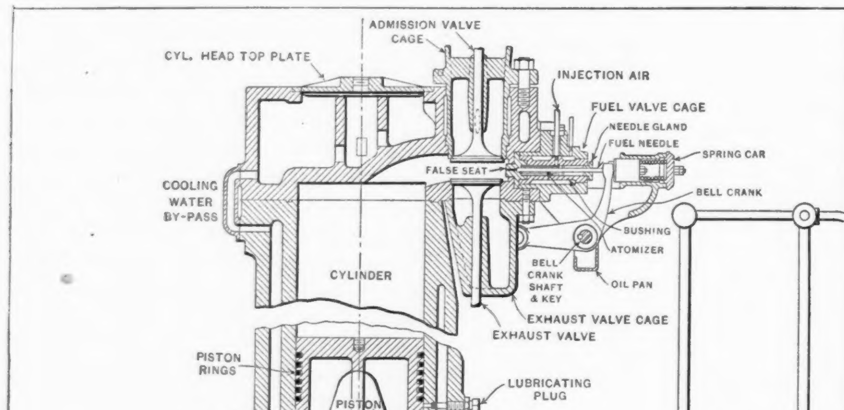


Fig. 1. Sectional View of Typical Diesel Oil Engine

In spite of the fact that the name Diesel has spread over the entire world, but few engineers in fields other than combustion engineering have ever taken the time to thoroughly study the movement of the engine and to learn of its proved practical application and use to the engineering world. The purpose of this article is, therefore, to present an idea of what the Diesel engine is, and to show in what fields the greater possibilities of the engine lie.

This type of engine may be defined in a few words as follows: An internal combustion engine which takes its fuel—crude, lowest grade fuel oil, or the residues from oil refining—into the cylinders, raw, without any previous transforming, and there converts it into energy, exerting that energy directly on the crankshaft through pressure on the piston head, without any intermediaries, thus producing the simplest, most direct and economical operation.

Principle of Operation

The cycle of operations of the Diesel engine operating on the "four-stroke cycle," comprising the Diesel ignition compression is as follows:

Stroke 1. Admission. The piston travels down or out, allowing the cylinder to fill with pure, fresh air from the inlet valves.

Stroke 2. Compression. The piston travels up or in, compressing air in the cylinder. The compression heats the air so much that the oil discharged into it will ignite and burn.

Stroke 3. Combustion. The piston travels down or out. At the beginning of this stroke, when the crank is on dead center, the fuel valve opens and the fuel charge of oil is sprayed into the heated air of the cylinder by a jet of air separately compressed by a small compressor. The spraying extends over 12 per cent of the working stroke of the piston and the combustion is gradual, the resulting pressures being even and sustained and not explosive.

Stroke 4. Exhausting. When the piston reaches the lower or outer end of the cylinder on stroke 3, the exhaust valve is opened, the pressure relieved and the piston travels up or in, driving out the exhaust gases of combustion.

Large Diesel Engines

Owing to the steady application of its power, the lack of vibration, comparative noiselessness, reliability, and to the small space which it occupies, the Diesel engine has become an almost ideal engine for marine work where oil may be had at nominal prices. An idea of the ease with which a Diesel engine may be controlled may be had from experiments run with the British motor ship *Eavestone*. This vessel is a ship of 4310 tons displacement recently built by Sir Raylton Dixon Co., Ltd. The 800 horsepower Diesel engine was largely built and installed by Richardsons, Westgarth & Co., Ltd., of Harthpool. She was capable of doing somewhat over nine knots. The action of the engines was very remarkable; they were reversed from full ahead to actually running astern in from nine to ten seconds, and this without any haste; in fact, with intentional deliberation. It was stated that the reversal had actually been completed in the remarkable time of six seconds. Observations showed that it only took about three seconds from the ringing of the telegraph till the engine was actually running ahead, the gear being already in the "ahead" position. Units of this character have been built for marine service up to about 2000 horsepower, single-acting type. The double-acting types are as yet but in the experimental stage.

Established Popularity of the Small Diesel Engine

The small and medium Diesel engine in sizes of from 40 or 50 horsepower in single-cylinder units, up to about 600 horsepower in four-cylinder units plays a vastly more important part in the commercial world than do the larger units; it has unquestionably come to stay, and has reached a high

* Abstract of an article in the January number of the Sibley "Journal of Engineering."

degree of perfection which places it right in line with the corresponding steam or gas engine plant as far as reliability and cost of operation are concerned, and far ahead of its competitors when considered from the standpoint of fuel economy. It is amazing to note how many manufacturers of gas and steam engines have taken up the manufacture of Diesel engines, because they found that the sale of suction-gas producer plants and smaller steam engines has fallen off alarmingly within the last few years.

The reason is plain. The single-acting four-cycle, single or multi-cylinder Diesel engine, but particularly the former, is comparatively simple in construction and operation. It does not require upkeep and attendance of boilers or gas

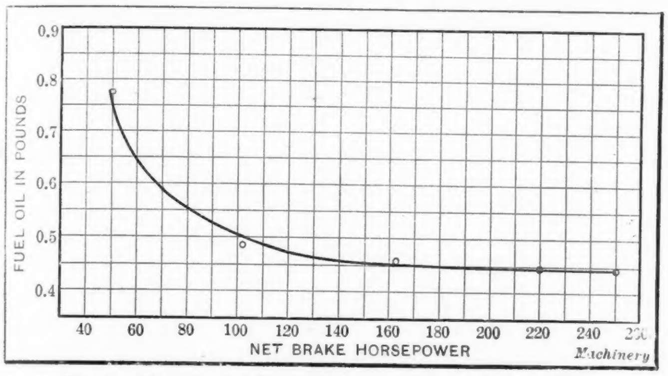


Fig. 2. Diagram showing Fuel Consumption of Diesel Engine

producers, and its cost, compared with that of a steam or gas plant, is reasonable. It can be installed in the basements of buildings below occupied dwellings. One of the greatest advantages, however, is the fact that the actual fuel consumption of Diesel engines taken over long periods of operation does not materially exceed the guaranteed figures; whereas, in gas producer and steam plants this excess is quite considerable. In a Diesel plant the human element—the skill of the operator—has much less influence upon the fuel economy than in a steam or producer gas plant, where everything depends upon the efficiency and intelligence of fireman and producer attendant.

On account of its low cost and great simplicity and variety of fuels which can be utilized the engines are being used and installed for various purposes, as for power transmissions, for operating flour mills, for lighting factories, office buildings, department stores, and for electrical works, etc. Lately they have been applied to the production of power where new and additional equipment is found necessary. It has been found that this engine can be very economically maintained as a reserve for relieving the original units of the peaks of the load and for like purposes.

Efficiency of the Diesel Engine

The efficiency of this type of engine and its characteristics may be readily obtained from the results of a test on a typical Diesel engine doing actual work. This test was conducted by A. C. Scott at the Scott Engineering Co., Dallas, Texas, on a 225 horsepower Diesel engine; the following data have been obtained which will give an excellent idea of the applicability of this engine to the generation of power.

The engine tested operates on the four-stroke cycle Diesel principle. It is a vertical three-cylinder unit, rated at 225 net brake horsepower, with allowance for power necessary to drive auxiliaries. It is directly connected to a Fort Wayne 200 K. V. A., three-phase, sixty-cycle, 2300-volt generator of 164 R. P. M. The auxiliaries consist of one belt-driven three-stage air compressor; one motor for driving compressor, 25 horsepower; and one exciter, 10 K. W., speed 850 R. P. M., belt-driven from the engine.

In many cases the air compressors for the starting and injection of air for Diesel engines are belted from the crankshaft of the engine; or when the compressor is motor driven, the power is supplied from the generator. During the series of tests, the power for the compressor was furnished from an outside source. Therefore, in order to obtain the net available power developed by the engine, readings were taken of the power consumption of the motor driving the compressor and a deduction made from the kilowatt output of the generator. No allowance was made for the efficiency of the motor; so that with the compressor belted to the engine shaft, the fuel consumption per brake horsepower hour would have been less than shown in the results given.

Fuel Oil Used

To obtain the amount of fuel oil consumed by the engine, it was only necessary to weigh the oil fed to the fuel pump, where the amount of oil actually fed to the engine cylinders is automatically controlled by the governor, and varied according to load requirements. For weighing the oil a 15-gallon tank and platform scale were used, and readings were taken at ten-minute intervals. A funnel of sufficient capacity was installed in order that a small amount of oil might be stored ahead of the governor, and the weighing tank properly re-filled with oil.

The oil used for these tests was taken from regular stock in ordinary use in this engine. Samples of oil were taken periodically, and the analysis made upon the total combined samples. The analysis of the fuel oil is given below:

British Thermal Units................	18,986 per pound of oil
Specific Gravity, 25.5 degrees C. to 27 degrees C. (78 degrees F. to 81 degrees F.)	0.8531
Viscosity, 33.3 degrees C. (92 degrees F.).............................	1.63
Flash Point	143.6 degrees F.
Burning (Fire) Point................	181.4 degrees F.
Sulphur	0.2 per cent
Water	Trace
Free Acid	None

The cost of this oil was $1.22 per barrel of 42 gallons, delivered; or practically 2.9 cents per gallon.

The instruments used in making the tests were all calibrated before using and due corrections made in the figures obtained. A water rheostat was found entirely satisfactory for the adjustment of all loads.

Tests

Six tests, each of three hours duration, were made with load changed for each hour period as follows:

Test Nos.	1	2	3	4	5	6
Net B. H. P.	2.25	49.7	111.39	162.97	219.63	245.62

Fuel Consumption

The fuel consumption for these various loads was in each instance obtained from an average of the readings taken at ten-minute intervals on the generator and the motor, and the records of oil used also taken every ten minutes. Fig. 2 is a curve showing in pounds the fuel oil consumed per net B. H. P. hour at the various loads. Test No. 1 is not included in this curve, as it was practically a no-load test. This curve was plotted by taking the average readings of output of the generator at the switchboard, corrected for the previously ascertained generator efficiency at the given load. From this was subtracted the actual kilowatt in-put to the motor and the resulting figure reduced to brake horsepower. It will be noted from the above curve, that the consumption of oil per brake horsepower hour increases but very little when the output is decreased from full load to about half load. At full load the fuel consumed was 0.441 pound per brake horsepower hour, or about 6.2 gallons per hundred net brake horsepower hours. When running at practically half load, the fuel consumed was 0.482 pound per brake horsepower hour, or about 6.8 gallons per hundred net brake horsepower hours. At quarter load the engine consumed 10.8 gallons of oil per hundred net brake horsepower hours.

* * *

The largest double-acting pile driver ever built was used for driving concrete piles for the Intercolonial Railway's new pier and shed at Halifax, Nova Scotia. The combined weight of the hammer with follower and follower guide is 24,000 pounds; weight of the ram only, 4000 pounds. The diameter of the cylinder is 14 inches and stroke 36 inches. The hammer drove 1800 reinforced concrete piles 24 by 24 inches square and 37 to 77 feet long.

RATIONAL METHODS IN ENGINEERING EDUCATION*

The work in machine design and construction at the Michigan Agricultural College is under one head. This permits of a superior selection and correlation of subjects. In this way the student, during the last three years of his course in mechanical engineering, has presented to him the fullest possible variety of subjects for design, and the work in the shops and the methods in design are more successfully correlated.

Many men, among others Mr. F. W. Taylor, have objected to the ordinary engineering graduate because of his lack of responsibility, perception and individuality. The endeavor at the college, the work of which is to be described, is to link good commercial practice with theory. The object is to put more seriousness into the teaching of engineering. School terms, such as "plate," are not used in the designation of drawings. The drafting-rooms are real drafting-rooms carried on along lines obtaining in good commercial practice. The main effort is concentrated on making the students more responsible men, and developing individuality and judgment. An endeavor is made to create a more businesslike atmosphere in the college, but the object is not to make engineering less rigid, but more interesting, fascinating and hence more understandable and better assimilated.

These things are done by applying principles as soon as possible after they have been enunciated. The examples are made real, doing away almost entirely with exercise pieces as such, both in the shops and in the drafting-rooms, and designing and building real machines in real ways for real uses. The great variety of machines and tools now being made in the shops permits of selecting in pedagogical order the proper sequence of exercises.

A need on the part of the students is created in exciting their desires by presenting subjects to them, first functionally and then following this by the pure theory underlying the subject at hand. They are led by means of that which is familiar to them, to that which to them has been unknown. The methods of making machines and tools follow the best practice; both processes, that of "building," and that of "manufacturing" in lots and by the use of special tools, jigs, etc., are used. The work differs from the condition in commercial shops only in the time required to produce a machine. The pedagogical order of selection of exercises does not permit of finishing a machine within a definite period. Of course, all of the equipment is not made at the college. The policy is to buy the very best machinery required for very accurate work, as for example, universal milling machines, and to make the general equipment.

In general, seniors and sometimes juniors design the machinery made. The details for the most part are worked out by sophomores. This follows the order in which work is done in the commercial drafting-rooms and it correlates well with the sophomore work in empirical machine design. The problems are attacked just as they would be in the commercial drafting-office. The need is first considered; a general specification sheet is then prepared by the instructor; the specifications are then elaborated by the student in accordance with his individual ideas, after which the final specification is checked and approved. As far as possible the students, in the senior year especially, work on different problems or on the same problem with different specifications. Drafting-room standards have been adopted which are rigidly followed during the three years that the students are engaged in the machine design. This follows good commercial practice.

In connection with the courses in machine design, the shop courses and the course in works management, a series of what may be termed "specialized inspection trips," has been adopted. These are not entertainment trips; nor are the students bewildered by examining all the processes occurring in any one factory. A need is first established such as, for instance, in machine tool design; after the students have endeavored to complete their specifications for, say, a radial drill by reference to a rather complete catalogue file, they find themselves somewhat shaky regarding some of the details. A trip to Lansing is then taken and radial drills, nothing else, are inspected, several different shops being visited.

A course is given in jig and fixture design, not with the idea of making tool designers of the graduates, but to acquaint them with the principles of this most important feature of manufacture. In this course especial emphasis is laid upon working hand in hand with the shop, conferences being held by the student with the machine shop foreman after the student has carefully studied his problem and has formed an opinion as to the best design, but before the work is done on paper. The jigs and fixtures designed are constructed in the shops and used in making the machinery. The courses in design and that in the construction of jigs are carried on simultaneously. The course in works management is also given during the same term, thus correlating three courses that go well together.

The course in works management includes such subjects as location and design of works, organization, costs, wages, labor, etc. The principles are given first and this is followed by a series of lectures on their application in the different departments of a factory. One advantage in giving these applications is to show to the student the relations, duties and limits of these several departments toward one another and to better enable him to fit himself into a commercial plant after graduation. In this course modern methods of production are carefully considered, and the course has enabled a number of the graduates to accept and hold positions along this line.

Efficiency is not only preached but also practiced. Indeed, the great variety and considerable number of machines and fixtures now going through the shops requires some system of checking and following up work, and this is done along approved commercial lines, using order blanks, etc., as in a first-class machine manufacturing plant.

Let us consider a hypothetical case: All orders for machines or tools that are to be made in the shops are issued, in triplicate, from the office of machine design and construction. The drawing numbers are also given here, following a standard system. The order is first issued to the pattern shops, two copies of the orders being sent with the blueprint, and the original being filed in the office under the head of "Orders issued." If the order is urgent, the pattern, when finished, is delivered to the foundry, otherwise to the pattern storage. The pattern number is given by the foreman of the pattern shop, who also has charge of the pattern storage department. The numbering system adopted at once locates the piece on its shelf. The pattern number is marked in red pencil in its proper place on the bill of material by the foreman of the pattern shop and the blueprint returned to the office with the one copy of the production order which has received the signature of the one to whom the pattern was delivered and filed under the head of "Orders filled." The second copy of the order is retained by the pattern shop. The pattern number is then put on the tracing and the casting production order is issued.

A "work order" card is also used, by means of which the shop can keep a record of the students' work, waste and efficiency, and the student is able to compare his time with that of a mechanic under similar conditions in a commercial shop. He can also obtain the shop cost of the work done, and thus have an idea of production values. The cost value of an automatic machine or special tool, such as a jig, and the rate at which the pieces to be machined could be turned out with the refined equipment, the time cost of the operation, etc., are studied. The student is thus enabled to study production while producing, and without expenditure of time.

Some of the machines now under construction are: six 14-inch engine lathes; a number of pattern lathes; a planer; a keyway slotter; two one-ton pneumatic hoists; a triplex power pump; a disk grinder; a steam hammer; a punch and shear; a star feed for a boring mill; and a vertical slotting attachment for the universal milling machine, and a great number of jigs, tools, and fixtures.

* Abstract of a paper read by Prof. Edward J. Kunze, assistant professor of mechanical engineering, Michigan Agricultural College, East Lansing, Mich., before the Michigan Engineering Society.

INVENTING MACHINES TO MAKE INVENTIONS MARKETABLE

SOME INTERESTING FACTS ABOUT INVENTIONS AND A FEW POINTS THAT THE INVENTOR USUALLY OVERLOOKS

BY E. R. MINER*

Probably ninety-eight out of every hundred men have, at times, dreams of becoming great inventors. Such dreams are usually colored by visions of some epoch-making discovery which will bring both fame and fortune. As a matter of fact, and according to history, the epoch-making discoveries have made comparatively few fortunes for the inventors. The fortunes have come to others after a long period of improvement, elimination and practical trial. Where there is one Alexander Graham Bell or Thomas Alva Edison, there are a thousand unknown and unremembered inventors. Inventors, as the old saying has it, may be born, but successful inventions are matters of pure business, the gradual evolution of new dresses for old ideas, the working out of new methods, new goods, and new applications to meet recognized trade requirements.

Whereas the large fortunes which have been made by some great and timely discovery or invention can be counted on the

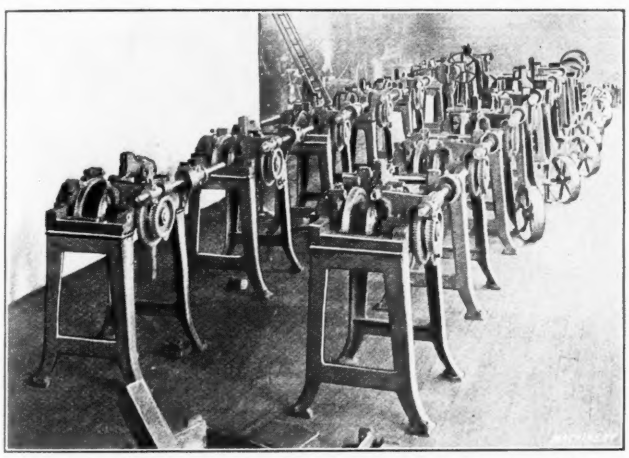


Fig. 1. A Battery of Sixteen Hair-pin Machines on the Assembling Floor in the Factory where built

fingers, the moderate fortunes which have been made by inventing some small article of practical and everyday use are not only numerous, but are well represented in all sections of the country. To merely invent something is not nearly as hard as knowing what to invent. Nearly any mechanic, if turned loose in a shop, could manage to invent things, but it would only be the occasional and exceptional man who would invent things which had a money value. The really new things or the radical changes are often the ideas of persons

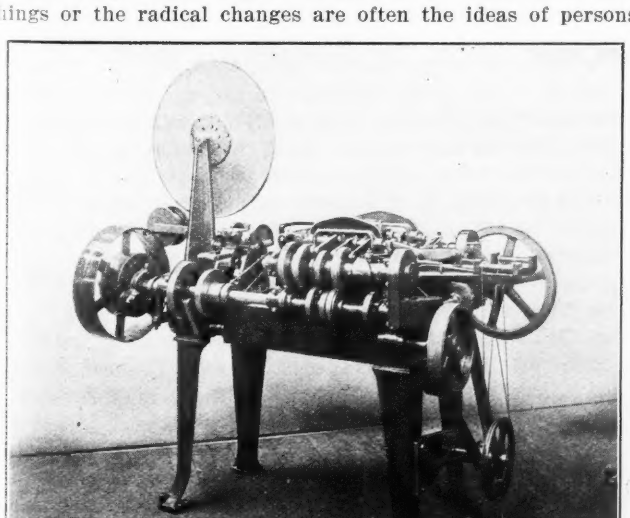


Fig. 2. A Pin Ticket Machine which prints the Ticket and makes and attaches the Pin for Tickets used on Clothing, etc., in Stores

who have but a surface knowledge of the business to which their invention applies. Not being bound by custom or by recognized methods or previous experience, they draw upon their imagination or inventive faculty. Lack of practical knowledge often does give that twist to the imagination which results in the new idea, but usually there must be a long

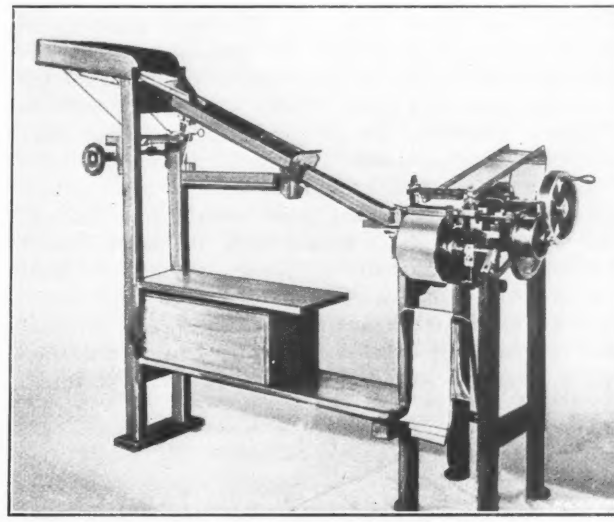


Fig. 3. Pin-sticking Machine which counts the Pins and sticks them into Papers

period of experiment and the bringing into play of practical knowledge to make the idea successful.

An electric lineman might be employed for twenty years in stringing heavy electric cable, yet his duties seem so commonplace that he never suspects or thinks that a large business could be created in making the clips or hooks which fasten the heavy cable to the supporting wire. The stranger, or possibly the young engineer who has charge of the line gang,

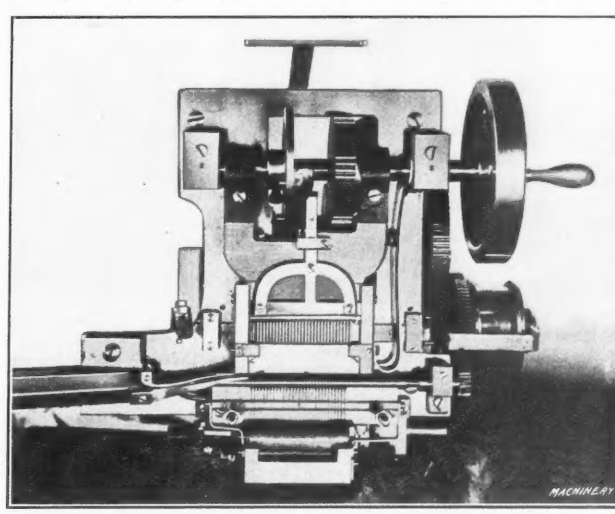


Fig. 4. Top View of the Principal Part of the Pin-sticking Machine shown in Fig. 3

being more or less unfamiliar with the work, notices the great number of clips or supports used. He also notices that the lineman cuts off a section of wire for making these clips, and immediately the idea is born as to why these pieces could not be manufactured more cheaply, more uniformly, or better, by machinery.

To successfully follow out such an idea, it is important to understand trade conditions, to know what the possible market would be, what price they would have to be sold at, what saving in labor or cost such article would represent, and after designing a clip which would meet all requirements, to know that a machine could be made for manufacturing such a clip both rapidly and cheaply. The failures in the invention business have been mostly by reason of the inventor going ahead on some idea which appealed to him personally, but on which his actual information was very small.

So apparently insignificant a product as the hairpin illus-

* Address: Box 315, Bridgeport, Conn.

trates in a simple manner the development of an idea into an industry. Wild thorns, sharpened sticks and shaped pieces of bone or shell were the original hairpins. Later on, the goldsmiths turned out by hand various devices in the way of bands and pins for holding the hair. The inventor of the wire hairpin is unknown, but the original hairpin was probably a piece of wire bent over in the center to form two straight legs of equal length.

The practical man stepped in, and understanding the various deficiencies of the hairpin as it existed, began to make improvements until today there are dozens of patents covering the point on the pin, the crimping or waving of the wires, the general shape, and other features which presumably make the hairpin better for the purpose intended. From this development, there proceeded the development of machines for manufacturing them. A hairpin made by hand or one requiring considerable manual labor would be a hairpin of rather high cost. As a consequence, the sales would be restricted. With automatic machinery, the hairpin becomes an article where cost is reduced to a mere trifle over the cost of the raw material, and as a consequence, it becomes an article of everyday necessity. A modern hairpin machine, depending upon the size and kind of the pin, will turn out from 75 to 200 per minute. At least one modern hairpin manufacturing company making nothing but hairpins keeps 75 machines running continuously with a general average of 100 per minute or 6000 per hour for each machine.

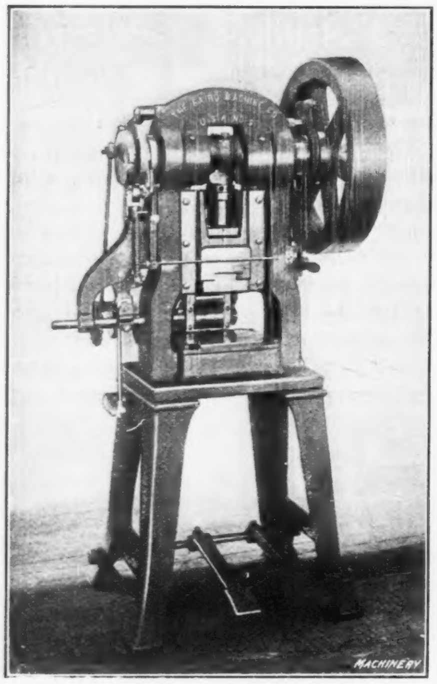


Fig. 5. A Roller Feed Automatic Press performing Half a Dozen Operations and provided with Automatic Stop

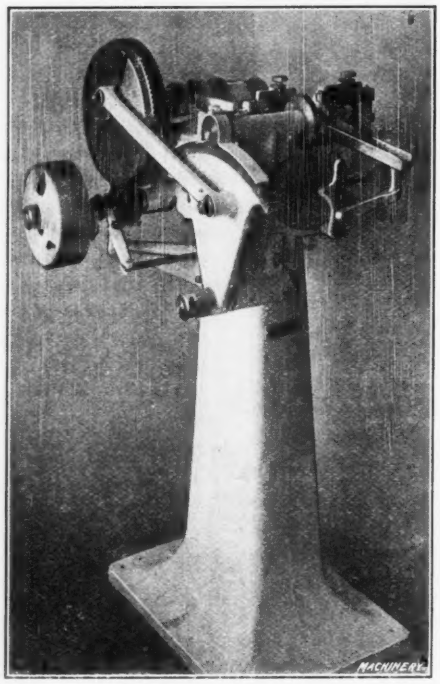


Fig. 6. A Spring Winding Machine which receives Wire from the Coil and completes Springs at the Rate of 150 per Minute

Fig. 7. Paper Clip Machine which turns out Finished Clips at the Rate of 450 per Minute

The improvements, the attachments, and the small changes that make a thing practical, make it conform to the requirements of the trade, and generally whip it into shape, are made by those who are thoroughly familiar with the business in hand and know what will be required by trade usage. There are very few inventions that ever have or ever can immediately revolutionize conditions, trade, methods, or things in general. A design or invention that is different beyond a certain point is held up to ridicule as a freak, and regardless of actual merit, it may be years before the public will accept such a design or invention at its true worth.

Wise manufacturers seldom put out freaks, but rather keep to their general design, making small changes here, and others there, until the trade is led to accept a freak as an outgrowth of gradual improvement. Few inventors see the reasonableness or business policy in this. The true inventor would immediately revolutionize business. He forgets the interests of those who have money tied up in a competing article, and would stuff his invention into the hands of every member of the trade. Admitting that an invention had real and superior merit, this, without previous education, is not a selling point. One shotgun might be so superior to another shotgun that there would be no comparison, and yet there would be no argument that would influence the man whose knowledge of shotgun requirements could not grasp the technical difference. To him they would both be shotguns.

To the general public, inventions that are radically new are things to let the other fellow fool with. This characteristic of the public will account for about one-half of the failures of inventors to put a really meritorious article on the market. The inventor, therefore, to be successful in a financial way, should be a business man. He should be broad in mind, and willing to see things as they exist, and not as he would have them. He must recognize that all things must be manufactured, and to be of any pecuniary benefit, they must be sold. To be sold, a product must adhere more or less to certain well defined standards. Such standards can be gradually altered, but they cannot be rushed at and immediately overthrown. In addition, to be manufactured at a cost which will admit of selling at a price which will be acceptable to the public and yet admit of a profit, special machinery or special applications of standard machinery may be necessary.

Successful invention (and by successful invention we mean invention which brings financial return to the inventor) is a business which requires close study of trade conditions and the possible demand brought about by modern improvement. The man who invents a garter clasp or a new type of hairpin, and can get them on the market in a proper manner, stands a better chance of being adequately rewarded than he who struggles for many years in an effort to build some type of a great power turbine.

There have been fortunes made on pins and other fortunes on hooks and eyes, but such fortunes have been built through a universal demand that called for quantity, and the popularity of these goods and immense sales for them have been created by reason of their very small retail cost. This low cost is made possible by automatic machinery that takes the wire or metal from a reel, feeds it through the machine, and drops out the completed article. Without such machinery, neither the pin business nor the hook and eye business would be possible, and the general public would still be using the makeshifts of our ancestors.

The inventor of the hook and eye or the inventor of the pin probably could not design a machine for making them, and they must perforce go to other inventors who could build one for them. Like thousands of other things, the hook and eye was an idea. Properly made and properly put on the

market, it was a builder of fortune, but it required the machinery back of the hook and eye to make such an idea successful.

The machine part of the proposition is really the fundamental basis for success. Fish-hooks would still be made by hand and at home by those who use them were it not for the automatic machines which turn them out so rapidly and at so low a cost that it would be a foolish waste of time and energy for anyone to attempt competition by older methods, even for their own use. The dollar watch is made possible through the design of special machinery. Even the automobile might still be an idea were it not for the machine builders who have made the rapid and uniform manufacture of the various parts possible.

Whether the article be a suspender buckle or a gas engine, the very first essential of operation will be "How can it be manufactured?" Financial success may entirely depend upon the answer to this one question. Throughout the New England states, that "Yankee district of the country so well known for its ingenious contrivances," there are shops which could be well called "inventors' factories." Probably the oldest of these, and perhaps the largest, is located in Bridgeport. To go through this factory is a liberal education on the

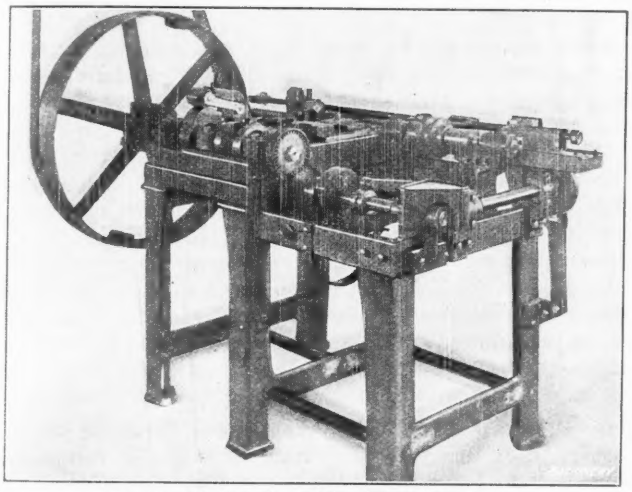


Fig. 8. Machine for making Hinges for Pianos

reduction of cost in manufacturing methods. The founder was the inventor of the original pin sticking machine, which counts the pins and sticks them in papers, so many pins to the row, and so many rows to the paper. A comparatively simple contrivance, it revolutionized the pin business, and is used in every country where pins are manufactured. It was the machine in this case that put into the hands of the manufacturers a popular, easily handled package for the retail sale, while at the same time making a reduction of cost in handling.

There is practically no industry today where the machines for making some part of the product are not the real factors of success. We hear of the inventor of a typewriter or of some other product, but we never hear of or give a thought to the inventor of special machines and special attachments which make the manufacture of the typewriter or product possible. And in the same way, there are manufacturers of special machinery whose factories are probably unknown to the general business man, yet special machines of their design are in every corner of the world, and turning out completed products or parts for pretty nearly every line of business.

A recent visit to one of these factories showed on the testing floor, having a final try-out before shipment, the following machinery: An upholstery tack machine, a form of press which pulled wire off a reel on one side, flat metal from a coil on the other side, and delivered large round headed upholstery tacks at the rate of 125 per minute; a patent thumbtack machine biting pieces out of a coil of flat brass and dropping those aids to the drawing board at 135 per minute; a butt hinge machine rolling off hinges faster than one would want to stand and count them; an electric lamp socket machine assembling the brass and porcelain sections at the rate of 60 per minute; a hook and eye machine spinning wire off a coil and dropping the familiar hump hook at 225 per minute; a paper clip machine making 300 clips per minute; a hairpin machine; a machine for making screw-on jar tops; a machine for making the wire guards for lanterns; a machine for making wire clothespins; a machine for making bed springs; a machine for making small springs; an eyelet machine for glove fasteners; a safety pin machine; a pin ticket machine for clothing; a machine for making, printing, and coating with paraffine the paper caps for milk bottles; a capping machine for beer bottles; a bucket machine; a garter clasp machine; and a great variety of wire and metal bending and forming machines.

Often a single machine will be found adequate to supply a whole industry. At the speed of 100 per minute, there is a total of 50,000 to 60,000 pieces to represent a ten-hour day. Now, 60,000 pieces per day for 300 days in the year often represents a greater quantity than can be disposed of, and there are cases where one machine pays good dividends, and is run perhaps only half or quarter of the time.

Work which is done by hand represents labor charges. It, also, if fairly large quantities are wanted, represents lack of uniformity and interchangeability. High labor charges require a high selling price, and the higher the price, the more restricted the field. Bring machinery into the question, and the labor charges for a single piece drop, the selling price drops with it, the selling field is enlarged, and the greater quantity demanded increases the business and produces a larger net profit. The machine or manufacturing part of a proposition is often lost sight of by reason of the ingenuity shown in a device, or its evident salability. Wrecked hopes are the reward of such short sightedness. The world may be waiting for a device to accomplish a certain purpose, but it will continue to wait if the price is beyond certain set limits.

There are hundreds of men earning a most excellent income by devising practical improvements for everyday products. These men do not always know all the angles regarding the particular product they may be working on, but they waste but little time on developing an idea until they know the probable market price, the quantity likely to be sold, and that it can be manufactured for the price by machinery.

* * *

REWARD FOR A RUBBER TIRE SUBSTITUTE

The Austrian War Department has offered a prize of $10,000 to be awarded to the person who with adherence to certain prescribed conditions will construct an elastic tire for motor trucks which possesses essentially greater durability, or, with equal durability, the attribute of essentially smaller cost of construction than the rubber tire, having, in addition, the properties of elasticity and adhesiveness of pure rubber. The weight must not exceed that of a pure rubber tire. Competitors must hand in a model of the fabric in natural or reduced size, together with drawing and description, before June 30, 1914, to the Automobil-Versuchsabeteilung (Automobile Trial Division), VI., Gumpendorferstrasse 1, Vienna, Austria. Further details may be found in the *Militarische Rundschau*, published at I. Graben 23, Vienna, and to be had also on application, in German, to the K. K. Kriegsministerium, Vienna, Austria.

* * *

A tap is a tool for cutting internal threads. Its efficiency does not depend on a highly polished shank, nor should its sale be affected by the characteristics of superficial finish. The fact is, though, that tap makers generally bestow a great deal of care and labor on superficial finish because of the prejudices of the average run of customers. A maker of forged pipe taps—having forged flutes and squares—leaves the flutes and square black after hardening but imitates the shape of milled flutes in order that purchasers shall not be adversely influenced by the too obvious fact that they are forged. As a matter of fact a forged flute tap should be better than when cut from the solid.

NOTES AND COMMENT

It is reported that Prof. Lummer, of Breslau, Germany, has discovered a method of obtaining carbon in a liquid condition. Leading scientists believe that the possibility of producing liquid carbon will also make it possible to produce artificial diamonds by its crystallization.

The Brown-Boveri Works, in Switzerland, are constructing what is claimed to be the largest electro-steam turbine set built. The group is of about 30,000 kilowatt and is intended for the Mark electric station at Westphalia. It is expected that steam turbines in single units of as high as 50,000 or 60,000 horsepower will be constructed within the near future.

It is stated in *Engineering* that sample pieces of cast-iron pipe that have been taken up after having been in use from sixty to seventy years have been found to be practically as good as new. Many cast-iron piping systems are in use in England that were laid eighty years ago. There are some cases known where cast-iron pipes have been in service for 250 years.

The postoffice and post roads committee of the House of Representatives, on December 8, approved a bill appropriating $100,000 of the annual postoffice budget. This will be used for purchasing a number of postal cars to be operated as an experiment from which to determine the advisability of owning and operating all such cars, instead of renting them from the railroads.

The longest stretch of railway in the world without a curve, according to *Railroad Men*, is in New Zealand, where there is a distance of 136 miles which extends in a perfectly straight line. This is remarkable in view of the fact that New Zealand is one of the most difficult countries in the world for railroad construction, as it is very mountainous, thus necessitating sharp curves and very heavy grades.

The gas-engine manufacturing business is reported to be in a flourishing state in Germany. The largest of the companies has a capital of 20,000,000 marks ($5,225,000) and paid a dividend of nine per cent on this capital during the past year. The total value of the manufacturing during the year was $6,190,000. Wages continue to increase, the total increase per man per day since 1906 amounting to over forty-five per cent.

The Russian aviator and aeroplane designer, Sikorski, has built a giant biplane weighing about 7000 pounds. The new machine, it is said, is propelled by four motors of 100 horsepower each, and the biplane is claimed to have a carrying capacity of twenty people, cabins with sleeping accommodations being provided. On the trial trip, the machine rose into the air after a run along the ground of 100 yards and circled the grounds without difficulty.

Electric furnaces for the ordinary steel foundry are now being regularly built and are claimed to give good results, electric melting having many advantages from a metallurgical standpoint. The new electric furnaces are made as nearly automatic in action as possible and many improvements in their design have been introduced during the last few years, so that it is claimed to be easier at present to get good results from an electric furnace than from the ordinary type.

Lomax is a planned city in Henderson, Ill., which will have some novel and interesting features for manufacturers. The founders have planned for shipping facilities, power, labor and living conditions. They propose to give each manufacturing plant free power for at least ninety years on the basis of one horsepower for each male employe, charging for additional power at the rate of twenty dollars per horsepower for a year. Workmen's houses will be rented on the basis of two per cent of the actual cost and the workman will be given a chance to buy and build on small reservations without cash.

An English method of treating metal case bullets and projectiles so that the bore of the rifle will become, after continued firing, even more highly polished than a new barrel, is described as follows: Bullets or the driving bands of projectiles are first roughed by means of sand-blast. They are then treated with sodium silicate (water glass) dissolved in water. In this solution kaolin is also suspended. After the water glass has dried on the bullets, they are given a protective coating of paraffine or shellac varnish and, in addition, the bullets are given a thin coating of a lubricant such as beeswax.

The Institution of Mechanical Engineers of Great Britain has instituted a fund for the purpose of giving financial aid to prominent members who, through no fault of their own, are in poor circumstances. About $20,000 has already been contributed to this fund. The idea is to keep the fund invested and to use only the income for relief. Engineers who have held high rank in the profession and to whom the engineering field as a whole may be largely indebted may be so placed in their declining years that aid from a fund created by fellow members in the profession may be necessary and acceptable.

A new method of obtaining a black rust-proof finish on iron or steel has recently been patented by F. Richards of Coventry, England, (United States patent 1,069,903). The process can be applied to hardened steel because the work need not be heated above the boiling point of water as is necessary with some of the other black-finish processes. A solution is made of 120 gallons of water, 3 pounds of manganese-dioxide, and one-half pound of concentrated phosphoric acid. This solution is placed in a suitable receptacle and heated to the boiling point. The iron and steel articles which have previously been cleaned are placed in the solution for from thirty to ninety minutes, after which they are removed, wiped and oiled with linseed oil.

In a recent lecture given at London, England, by Mr. Richard Kerr, the title of "Father of Wireless Telegraphy" was given to James B. Lindsay, who was born in 1799 and died in 1862. This man, a not too-prosperous schoolmaster at Dundee Gaol, with a salary of £50 ($250) per year, made his own batteries and coils and sent wireless messages across the Tay and other Scottish rivers and lakes as well as across the Solent. He is said to have declared that if he only had the means to extend his experiments there was no reason why he could not send messages across the Atlantic. It was essential to the Lindsay system that there be a stretch of water between the transmitter and the receiver. Lindsay was a man of remarkable genius, but sometimes turned his energy to rather peculiar accomplishments. He died while engaged in the compilation of a dictionary on fifty languages, and was at that time regarded by his neighbors as "nutty."

Hydrogen compared to air in weight is near the irreducible minimum of the highest vacuum. There is little to hope for in reducing the bulk of dirigible balloons from the use of a much lighter gas. But an astronomer who makes the claim that the upper atmosphere of the earth is composed of coronium, a gas considerably lighter than hydrogen, suggests its use as a substitute for the latter in balloons. Inasmuch as the coronium gas is estimated to be mostly at a height of 135 miles, there is little prospect now of ever gathering it from its native "heath" in appreciable quantities. But if it could be obtained the gain would be small. At sea level and sixty degrees F. temperature, hydrogen gas weighs about 38 grains per cubic foot while air at the same pressure and temperature weighs 540 grains. The buoyancy of hydrogen is due to the difference between its weight and that of the same volume of air. The buoyancy of one cubic foot of hydrogen, then, under the given conditions is $540 - 38 = 502$ grains. If hydrogen could be replaced by a gas weighing one-fifth as much, the gain in buoyancy would be 30 grains per cubic foot, or only six per cent.

HOBBING VS. MILLING OF GEARS*

A COMPARATIVE STUDY OF THE TWO METHODS FROM THE POINT OF VIEW OF BOTH QUALITY AND QUANTITY

BY JOHN EDGAR†

The adverse criticism of the gear hobbing process has been the cause of many interesting investigations, and one of the most important of these has been the comparative study of the condition of the surfaces produced by the hob and by the rotary cutter. In making such a comparative study, it is necessary that the investigator possess the required practical knowledge, and also that he be willing to admit a point, even though his favorite processes may suffer by the comparison.

Feed Marks produced by Rotating Milling Cutters

While both the gear hobbing machine and the automatic gear cutter use rotating cutting tools, the operations cannot be placed on a common basis and considered as similar milling operations, although they may, to a certain extent, be compared as such. In comparing the quality of the surfaces produced by the two processes, consider first the milled surface produced by an ordinary rotary cutter. This surface has a series of hills and hollows at regular intervals, the spacing between these depending upon the feed per revolution of the cutter, and the depth on both the feed and the diameter of the cutter. The ridges are more prominent when coarse feeds and small diameter cutters are used. These feed marks are the result of the convex path of the cutting edge and the slight running out of the cutter, which is inevitable in all rotary cutters with a number of teeth. As is well known to those familiar with milling operations, the spacing of the marks does not depend on the number of teeth in the cutter. Theoretically, it should depend on this number, but as it is practically impossible to get a cutter which will run absolutely true with the axis of rotation, only one mark is produced for each revolution, and, hence, the spacing becomes equal to the feed per revolution. The eccentricity of the cutter with the axis of rotation is, therefore, the factor which, together with the diameter of the cutter and the feed per revolution, determines the quality of the surface, other conditions being equal.

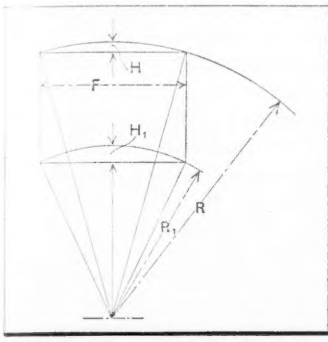


Fig. 1. Diagram illustrating the Relation between Feed, Diameter of Cutter and Depth of Feed Marks

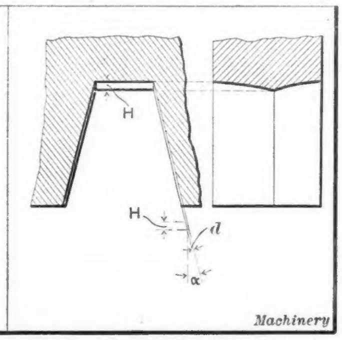


Fig. 2. Diagram for finding Depth of Feed Marks on Side of Tooth cut by Milling Cutter

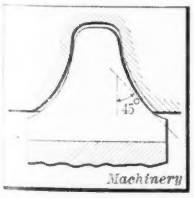


Fig. 3. Angle which limits the Feed

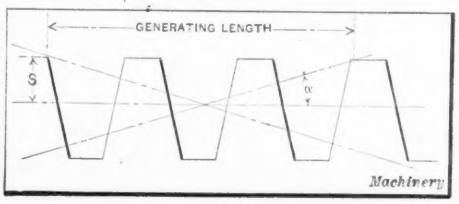


Fig. 4. Diagram for deducing Formulas for analyzing Action in Gear Hobbing Machine

The depth of the hollow produced by the high side of the revolving cutter is equal to the height or rise of a circular arc, the radius of which equals the radius of the cutter, and the chord of which equals the feed per revolution. (See Fig. 1). The length of the chord or the feed per revolution may be expressed:

$$F = 2 \times \sqrt{2\,HR - H^2}$$

in which F = feed per revolution;
H = height of arc;
R = radius of cutter.

Since H^2 is a very small quantity, it may be discarded in the expression, which is then simplified to read:

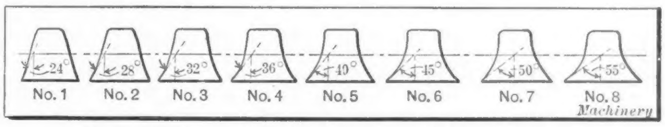


Fig. 5. Angles limiting the Feed for 14½-degree Standard Gear Cutters

$$F = 2 \times \sqrt{HD}$$

in which D = diameter of cutter.

Transposing this expression, we obtain $H = \frac{F^2}{4D}$, which is an approximately correct expression of the depth of the hollows produced by milling. As an example, take an 8-pitch rack cutter, with straight rack-shaped sides, 3 inches in diameter, milling with a feed per revolution of 0.1 inch. The depth of the feed marks at the bottom of the cut will be equal to:

$$\frac{(0.1)^2}{4 \times 3} = 0.00083 \text{ inch.}$$

The working surface of the tooth, however, is produced by the side of the cutter, as illustrated in Fig. 2, and the depth of the feed marks is normal to the surface, and is expressed as:

$$d = H \times \sin \alpha$$

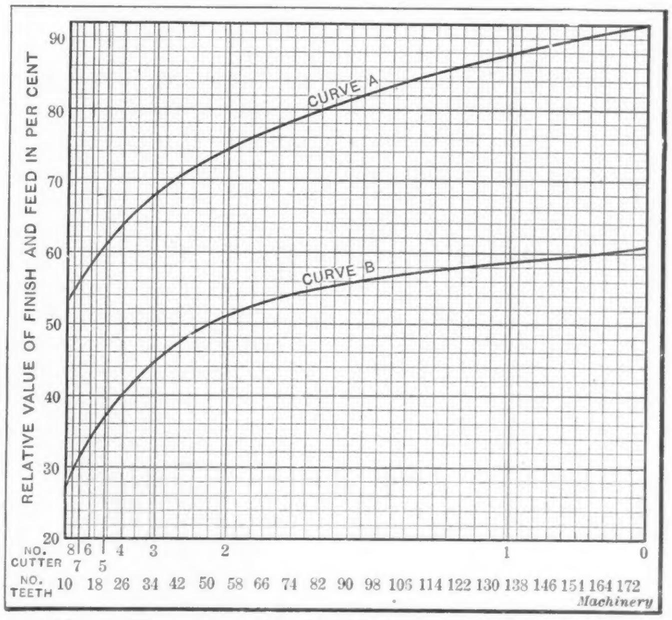


Fig. 6. Diagrams showing the Relation between Feed, Finish, and Number of Teeth when cutting Gears with Formed Gear Cutters

in which d = depth of the feed marks on the side of the tooth, and α the angle of obliquity. In the example given, the depth d would equal 0.00021 inch, for a 14½-degree involute tooth.

The depth of the feed marks is inversely proportional to the diameter of the cutter, and is, therefore, greater at the point of the tooth than at the root. In the example given, the depth would be 0.00025 inch at the extreme point of the rack tooth. It is thus apparent that the quality of the surface at any position along the tooth from the root to the point depends upon the diameter and form of the cutter and the feed per revolution.

In Fig. 3 is shown the outline of a No. 6 standard 14½-degree involute gear cutter. This outline, at the point close

* See also MACHINERY, July, 1912, engineering edition, "Hobs for Spur and Spiral Gears," and the articles there referred to.
† Address: 61 Bruce Ave., Windsor, Ontario, Canada.

to the end of the tooth of the gear, is a tangent inclined at an angle of 45 degrees, as indicated. Hence, the depth of the revolution marks is:

$$\frac{(0.1)^2}{4 \times 2.5} \times \sin 45° = 0.000707 \text{ inch, instead of } 0.00024 \text{ inch, as}$$

in the case of the straight rack tooth. It is evident that to produce an equal degree of finish with that left by the rack cutter, the feed must be considerably less for a No. 6 involute gear cutter than for the rack cutter. In Fig. 5 is shown the full range of cutter profiles from Nos. 1 to 8, with the angle of the tangent in each case which determines the quality of the surface under equal conditions of feed and diameter of cutter.

If the depth of the feed marks is used as the determining factor in comparing the condition of the surfaces produced by a series of cutters, it is evident that if the surface produced by the rack cutter is taken as a standard, the feed for cutting a pinion must be considerably less than the feed used for cutting gears with a large number of teeth. In fact, if a rack cutter is fed 0.100 inch per revolution, a No. 8 standard involute gear cutter should not be fed more than 0.055 inch per revolution to produce an equally good surface. The feed is proportional to the square root of the reciprocal of the sine of the angle of the limiting tangent.

If we assume the accuracy of the surface left by the straight-sided rack cutter as equal to 100 per cent, then the relative feeds required for cutting gears with any formed cutter can be calculated. This has been done, and the results are shown plotted in curve *A*, in Fig. 6. This curve is based on an equal depth of the feed marks for the full range of numbers of teeth in the gears. If, on the other hand, the surfaces left by the cutter for a given feed per revolution are compared, the depth of the feed marks will vary with the sine of the angle of the limiting tangent, and taking the straight-sided rack cutter as a basis, the relative accuracy of the surfaces is inversely proportional to the sine of the angle, and is plotted in curve *B*, in Fig. 6.

Comparison between Surfaces produced by Milling and Hobbing

A relation has now been established between the quality of the surface and the permissible feeds for cutters for cutting gears with any number of teeth. We will now consider the condition of the surface produced by a hob in a gear hobbing machine. The hob is made with straight-sided rack-shaped teeth and with sides of a constant angle, and is used

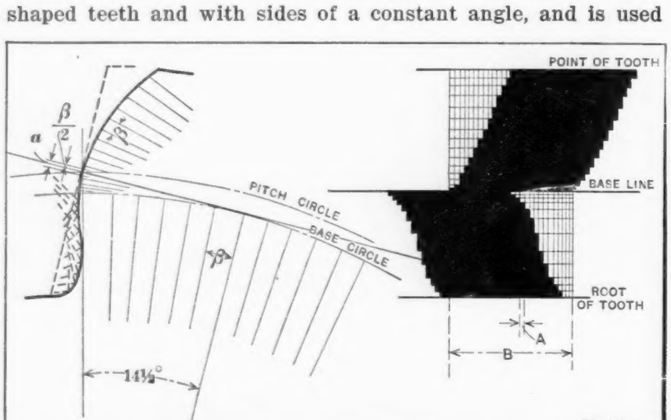


Fig. 7. Relative Width and Position of Flats produced by Gear Hobbing Machines. A indicates Feed per Each Generating Tooth; B, Feed per Revolution of Blank

to produce gears with any number of teeth. We may therefore assume that it is cutting under the conditions governing the rack cutter, as just explained, the surface produced being considered merely as a milled surface. If this assumption be correct, then the quality of the surface produced by a hob, whether cutting a gear of twelve teeth or of two hundred teeth, will be the same for a given feed, and the same relation exists between the hob and any formed cutter that exists between the rack cutter and any formed cutter; hence, curves *A* and *B*, in Fig. 6, may be assumed to show the permissible feeds and the quality of the surfaces produced by formed cutters when compared with the surfaces produced by a hob, provided the surfaces are considered merely as milled surfaces. However, a condition enters in the case of the hob which has no equivalent in the case of the formed milling cutter, and this influences the condition of the surface. This condition is the distortion of the hob teeth in hardening which causes them to mar the surface of the tooth by "side swiping," producing a rough surface. The eccentricity of the hob with the axis of rotation also has a different effect on the surface than in the case of a formed gear cutter. The effect is shown in a series of flats running parallel with the bottom of the tooth, if excessive; if the eccentricity is small, the effect will merely be to round the top of the tooth. These inaccuracies, however, can be taken care of in a number of ways.

Comparison of Output

For reasons not connected with the quality of the surface, the hob may be worked at a greater cutting speed and feed than a rotary cutter, when cutting from the solid, the reason being due to the generating action of the hob which results in the breaking of the chips. This preserves the cutting edges and reduces the heating effect of the cut, and explains why the hob may give such good results as compared with a rotary cutter in the matter of output. It is possible to get good results in the general run of work in the hobbing machine in one-third to one-half of the time required in an automatic gear cutter. The accompanying table gives the results obtained on automobile transmission gears with automatic gear cutting machines and hobbing machines. If anything, the conditions under which the comparisons were made favored the automatic machines. The speed of the cutter in all cases was 120 revolutions per minute, except in the case of the 13-tooth pinion, when the speed was raised to 160 R. P. M. to increase the output. The hob was run at a speed of 105 R. P. M., in all cases. The hob and cutters were of practically the same diameter. The results were obtained in producing an ordinary day's work and clearly indicate the advantage of the hobbing process over the milling process, when the quality of the tooth surface alone is considered, on the basis that both processes produce a milled surface.

COMPARISON OF TIME REQUIRED FOR CUTTING GEARS ON AUTOMOBILE GEAR CUTTING MACHINES AND HOBBING MACHINES

Number of Teeth	Automatic Gear Cutters		Gear Hobbing Machines	
	Feed, Inches	Time, Minutes	Feed, Inches	Time, Minutes
32	0.022	15	0.050	6.5
31	0 020	19	0.050	9
24	0.024	22	0.050	6
17	0.020	8	0.050	4
17	0.020	17.5	0.050	5
16	0.018	10.5	0.050	7
13	0.013	6.25	0.050	6

Machinery

The Tooth Outline

Going further into the subject, we will take up the question of the tooth outline. The tooth of a gear milled with an ordinary milling cutter must be, or at least is expected to be, a reproduction of the outline of the cutter, and since each cutter must cover a wide range of teeth, the outline is not theoretically correct, except for one given number of teeth in the range. Theoretically speaking, the outline of the hobbed tooth may be considered as a series of tangents, the tooth surface being composed of a series of flats parallel with the axis of the gear. To show the significance of these flats, assume, for example, that a gear with thirty-two teeth is cut with a standard hob, 8 pitch, 3 inches in diameter, having twelve flutes. The length of the portion of the hob that generates the tooth surface is $2\ S \div \tan\ \alpha$, where α is the pressure angle, as indicated in Fig. 4. The number of teeth following in the generating path is:

$$\left(\frac{2\ S}{\tan\ \alpha} \div \text{circular pitch}\right) \times \text{number of gashes.}$$

In this case, the generating length is approximately 0.96 inch, and there are thirty teeth in the generating path. The flats of those parts of the tooth outline which each of the hob teeth form vary in width along the curves. They are of

minimum width at the base line and of maximum width at the point of the tooth. The width of the flats at the pitch circle is proportional to the number of teeth in the gear, the number of gashes in the hob, and the pressure angle. The angle β, to the left in Fig. 7, which is the angle between each flat, is proportional to the number of teeth in the gear and the number of gashes in the hob. In the example given, it is:

$$\beta = \frac{360 \times \dfrac{0.96}{3.1416 \times 4}}{30} = 0.91 \text{ degree, or 55 minutes.}$$

The Width of Flat Produced

The width a of the flat at the pitch line is equal to twice the tangent of one-half β times the length of the pressure line between the point of tangency with the base line and the pitch point, and is:

$$a = 2 \tan \tfrac{1}{2} \beta \times \tan 14\tfrac{1}{2}° \times 2 = 0.0081 \text{ inch.}$$

This is not a flat that could cause serious trouble. As in the case of the feed marks, it is not the width of the flat alone that is to be considered, but the depth must be taken into account; in fact, the quality of the surface may be spoken of as the ratio of the depth to the length of the flat. The depth of the flat is the rise or height of the arc of the involute and is approximately proportional to the versed sine of the angle $\frac{1}{2}\beta$, and with the pitch assumed in the example given, would be 0.000015 inch. It is difficult to conceive of any shock caused by this flat, as the gear teeth roll over each other. The action of the hob and gear in relation to each other further modifies the flat by giving it a crowning or convex shape. In fact, the wider the flat the more it is crowned. This explains the fact that hobs with a few gashes produce teeth of practically as good shape as hobs with a large number of gashes. It is desirable, therefore, to use hobs with as few gashes as possible, because from a practical point of view the errors of workmanship and those caused by warping in hardening increase with the number of flutes.

A peculiar feature of the hobbed tooth surface is shown to the right in Fig. 7, which illustrates the path on a tooth produced by a hob in one revolution. In fact, there are two distinct paths, the first starting at the point of the tooth and working down to the base line, the cutting edges of the hob tooth then jumping to the root of the tooth and working up to the base line, producing the zigzag path shown.

Conclusion

That the flats so commonly seen in the results obtained from the hobbing machine are not due to any faults of the process that cannot be corrected, but are due to either carelessness on the part of the operator in setting up the machine without proper support to the work or to the poor condition of the hob or machine, and that nearly all cases of flats can be overcome by the use of a proper hob, may be assumed as a statement of facts. When the hobbing machine will not give good results, the hob is in nearly all cases at fault. If a gear is produced that bears hard on the point of the teeth, has a flat at the pitch line or at any point along the face of the tooth, do not think that the process is faulty in theory, or that the machine is not properly adjusted, or that the strain of the cut is causing undue torsion in the shafts, or that there is backlash between the gears in the train connecting the work and the hob; these things are not as likely to cause the trouble as is a faulty hob.

After an experience covering all makes of hobbing machines, the writer has come to the conclusion that the real cause of the trouble in nearly every case is a faulty hob. Machine after machine has been taken apart, overhauled and readjusted, and yet no better results have been obtained until a new and better hob was produced. The faults usually met with in hobs were referred to in an article, "Hobs for Spur and Spiral Gears," in MACHINERY, July, 1912. The means for getting the hob into a good working condition were explained in the same article. It is not desired in any way to disparage the formed cutter process in favor of the hobbing process, but simply to state the facts as they appear. In every case, practice seems to substantiate the conclusions arrived at.

CALCULATIONS FOR ROLLER CHAIN DRIVES*

BY G. M. BARTLETT†

There is much that is still to be learned about the action of a roller chain upon its sprockets when running at high speed. Systematically tabulated data bearing on the durability of chains under various conditions of linear velocity, angular velocity, chain pull and numbers of teeth on the sprockets are very meager, and no conclusions have yet been reached by which it is possible to predict the behavior of a given chain under unusual conditions. With a light, steady load, short pitch, and large sprockets, chains have run with perfect satisfaction at speeds as high as 4000 feet per minute; while with heavier loads, and sprockets with but few teeth, chains have been wrecked in a short time at speeds not exceeding 1200 feet per minute.

Under ordinary conditions, however, certain rules derived from experience can be used for chain drives where the speeds range from 400 to 1200 feet per minute. Within these limits the load that can safely be carried by the chain varies inversely as the chain speed. That is, if a chain can work safely under a tension of 400 pounds at a speed of 600 feet per minute, the safe working load at 1200 feet per minute would be only 200 pounds. The load that a chain can carry should be based upon the projected wearing surface of the rivet, and not upon the ultimate strength of the chain. To find the projected rivet area of a chain, multiply the rivet diameter by the length of the bushing. To find the approximate horsepower of a chain for speeds between 400 and 1200 feet per minute, multiply the projected rivet area by 24. This is based upon an allowable pressure of 1000 pounds per square inch of projected rivet area at a speed of 800 feet per minute and of 2000 pounds at 400 feet per minute. To select a chain for a given horsepower, divide the horsepower by 24. The result is the projected rivet area (approximately) of the required chain expressed in square inches.

PITCHES OF CHAINS FOR VARIOUS HORSEPOWERS

Pitch of Chain, Inches	Type of Chain	Horsepower at 400 to 1200 Feet per Minute	Pitch of Chain, Inches	Type of Chain	Horsepower at 400 to 1200 Feet per Minute
1	Solid blocks	½ to 2¼	1	Roller chain	4½ to 8
1	Twin roller chain	1⅛ to 2	1¼	Roller chain	7 to 12½
½	Roller chain	1⅛ to 2	1½	Roller chain	10⅛ to 18
⅝	Roller chain	1¾ to 3⅛	1¾	Roller chain	13¾ to 24½
¾	Roller chain	2½ to 4½	2	Roller chain	18 to 32
			......	....	....

Machinery

The accompanying table will be found useful in roughly determining the pitch of the chain to be used for a given horsepower. As chains of the same pitch are made of various widths and of various rivet diameters, the range in carrying capacity is rather large. The upper limit is the horsepower that can be transmitted by a chain of the greatest width and largest rivet diameter commonly supplied at the present time. Sprockets with less than 14 teeth should not be used except where the chain speed is low. The center distance, wherever possible, should be at least one and one-half times the diameter of the larger sprocket.

Having determined the pitch of the chain to be used, the pitch diameter, bottom diameter, and outside diameter of the sprockets can be found from the tables usually published by chain makers, or they may be calculated from the formulas:

$$\text{Pitch diameter} = \frac{P}{\sin \dfrac{180°}{N}} \qquad (1)$$

where N = number of teeth in sprocket and P = pitch of chain.

$$\text{Bottom diameter} = \text{pitch diameter} - \text{roller diameter} \qquad (2)$$

$$\text{Outside diameter} = \text{pitch diameter} + P\,(3N - 8) \div 5N \qquad (3)$$

* For further information on the subject of chain drives and allied subjects, see "Standard Sprockets for Detachable Link Belts," published in MACHINERY, August, 1913; "Lineometer for Determining Chain Lengths," May, 1912; "Design of Dished Sprockets," October, 1911; "Worm vs. Chain Drive for Auto Trucks," May, 1911; and "Chain Drives," February, 1909.

† Address: Diamond Chain & Mfg. Co., Indianapolis, Ind.

To estimate the outside diameter roughly, divide the number of teeth by 3 and multiply by the pitch.

To calculate the chain length, let

C = distance between centers;
D = pitch diameter of large sprocket;
d = pitch diameter of small sprocket;
N = number of teeth on large sprocket;
n = number of teeth on small sprocket;
P = pitch of chain;
θ = angle between straight part of chain and line of centers of sprockets.

$$\operatorname{Sin} \theta = \frac{D - d}{2C} \tag{4}$$

L = length of chain in inches

$$= \frac{P}{180}\left[N\,(90^\circ + \theta) + n\,(90^\circ - \theta)\right] + 2C \cos \theta. \tag{5}$$

This gives a theoretical chain length. The actual length must, in general, be enough greater than this to make it an even multiple of the pitch. If, however, a special offset link is used, the chain length will be an odd multiple of the pitch. This sometimes means that there will be a considerable sag in the chain, unless the center distance can be adjusted to take up the slack.

To calculate the center distance for a tight chain, let:

Z = chain length in pitches $= \frac{L}{P}$;

C_1 = the center distance for a tight chain, in inches.

$$C_1 = \frac{P}{8}\left[2Z - N - n + \sqrt{(2Z - N - n)^2 - 0.824\,(N - n)^2}\right] \tag{6}$$

This is an approximate formula, but it will be found sufficiently accurate for most cases. If greater accuracy is required, continue the calculations as follows:

$$\operatorname{Sin} \theta_1 = \frac{D - d}{2C_1} \tag{7}$$

$$C_2 = \frac{P\left[2Z - N - n - (N - n)\dfrac{\theta_1}{90}\right]}{4 \cos \theta_1} \tag{8}$$

This formula is exact, but the value of θ is only approximate, since it was determined indirectly from Formula (6). The value of C_2 is, however, more nearly correct than that of C_1, and one may obtain any desired degree of approximation by solving (7) with the value of C found in (8), and then solving (8) with the value of θ found in (7). One or two alternations will give a result more accurate than would ever be required in practice.

To calculate the chain velocity V in feet per minute, the chain tension T in pounds, and the horsepower $H.P.$, use the following formulas, in which S will represent the number of revolutions per minute:

$$V = \frac{SNP}{12} \text{ or } V = 0.262\ DS \text{ feet per minute} \tag{9}$$

where N = number of teeth on sprocket and D = diameter of sprocket in inches.

$$T = \frac{33{,}000 \times H.P.}{V} \text{ or } T = \frac{126{,}283 \times H.P.}{SD} \text{ pounds} \tag{10}$$

$$H.P. = \frac{VT}{33{,}000} \tag{11}$$

To find the minimum safe shaft diameter D_1 for a given sprocket:

$$D_1 = 3\sqrt[3]{\frac{H.P.}{S}} \tag{12}$$

This is for a shaft of mild steel subjected to a twisting moment only. For a shaft subjected to a combined twisting and bending moment, multiply the result of Formula (12) by 1.5. As an example, let it be required to select the chain and sprockets to transmit 15 horsepower from a motor running at 600 revolutions per minute to a lineshaft running at 230 revolutions per minute, the outside diameter of the sprocket on the lineshaft not to be greater than 20 inches, and the center distance to be approximately 40 inches. The projected rivet area of the chain should be not less than H. P. ÷ 24 or 0.625 square inch. A chain is found whose rivet diameter is 7/16 inch; length of bushing, 1.440 inch; projected rivet area, 0.629 square inch; pitch, 1½ inch; diameter of roller, ⅞ inch; width of roller, 1 inch; and ultimate strength, 21,000 pounds. Since the maximum outside diameter of the larger sprocket is 20 inches, the maximum number of teeth will be 40, as found in a table of sprocket diameters. But it is necessary for the number of teeth to be so selected that the velocity ratio will be as close to $\frac{230}{600}$ as possible. Dividing 230 by 600 we have 0.3833 as the decimal equivalent of the ratio. Referring to a table of decimal equivalents of gear ratios, such as may be found in MACHINERY'S Data Sheet No. 158 (published in September, 1912), the nearest fraction to 0.3833 with a denominator not greater than 40 is $\frac{5}{13}$ or $\frac{15}{39}$. We may therefore choose 15 and 39 as the numbers of teeth on the driving and driven sprockets, respectively.

$$\text{Chain velocity} = V = \frac{SNP}{12} = \frac{600 \times 15 \times 1.5}{12} = 1125 \text{ feet per minute.}$$

$$\text{Chain pull under full load} = T = \frac{33{,}000 \times 15}{1125} = 440 \text{ pounds.}$$

As a lineshaft is usually subjected to both twisting and bending stresses, the minimum allowable diameter would be

$$1.5 \times 3\sqrt[3]{\frac{H.P.}{S}} = 4.5\sqrt[3]{\frac{15}{230}} = 1.811 \text{ inch.}$$

The use of a 1⅞-inch shaft would be good practice.

The chain length is found from Formulas (4) and (5), thus:

$$\operatorname{Sin} \theta = \frac{18.641 - 7.215}{2 \times 40} = 0.142825$$

$$\theta = 8^\circ\ 12'\ 40'' = 8.2111^\circ \qquad \operatorname{Cos} \theta = 0.98975$$

$$L = \frac{1.5}{180}\left[39\,(90^\circ + 8.2111) + 15\,(90^\circ - 8.2111)\right] + 2 \times 40 \times 0.98975 = 121.321 \text{ inches.}$$

The next higher number than this which is an even multiple of the pitch is 123 inches. Hence $L = 123$ inches or 82 pitches. But if the center distance is 40 inches, there will be considerable sag in the slack portion of the chain. To take up this sag, the center distance must be increased to

$$C_1 = \frac{1.5}{8}\left[2 \times 82 - 39 - 15 + \sqrt{110^2 - 0.824\,(24)^2}\right] = 40.841 \text{ inches.}$$

If this is not thought sufficiently accurate, Formulas (7) and (8) may be used, thus:

$$\operatorname{Sin} \theta_1 = \frac{11.426}{2 \times 40.841} = 0.13988$$

$$\theta_1 = 8^\circ\ 2'\ 27'' = 8.0408^\circ$$

$$\operatorname{Cos} \theta_1 = 0.99013$$

$$C_2 = \frac{1.5}{4 \times 0.99013}\left[2 \times 82 - 39 - 15 - (39 - 15)\,\frac{8.0408}{90}\right] = 40.849 \text{ inches.}$$

This second approximation differs from the first by 0.008 inch. Substituting this value in (7) and (8), we have:

$$\operatorname{Sin} \theta_2 = \frac{11.426}{81.698} = 0.13985$$

$$\theta_2 = 8^\circ\ 2'\ 20'' = 8.0389 \qquad \operatorname{Cos} \theta_2 = 0.99016$$

$$C_3 = \frac{1.5\left(110 - 24 \times \dfrac{8.0389}{90}\right)}{4 \times 0.99016} = 40.848 \text{ inches.}$$

Further approximations would affect only the decimal places beyond the third. This shows that Formula (6) is sufficiently accurate for most practical purposes.

As a final word, it may be pointed out that the life of a chain drive can be greatly lengthened by enclosing it in a case and running it in oil. Also, that both the efficiency and wear will be improved by the use of a strut rod, or its equivalent, rigidly connecting the bearings of the two sprockets, thus preventing relative motion between the sprocket centers.

PUNCHING MACHINE FRAMES*

THEORY OF SHEAR AND PUNCH FRAME DESIGN BASED ON MODERN ANALYSIS OF THE STRESSES

BY C. D. ALBERT†

While numerous articles have appeared dealing with the applicability of the theory of curved bars to such curved members as crane hooks, punch and riveter frames, and the like, the dictates of the theory as to the proper outline for such members have seldom been touched upon.

Careful mathematical and experimental investigation has shown that to approximate the maximum induced stress in a steel or wrought iron crane hook the expression used must take account of the curvature of the member. In the discussion to follow, it will be assumed that the theory of curved bars may be extended to curved members of cast iron, so long as the maximum stress induced does not exceed one-third to one-fourth the ultimate tensile strength of the material. Such limits are imposed, since up to such limiting stresses the stress-strain diagrams for cast iron in tension and in compression are almost straight and practically coincide. It can therefore be assumed without appreciable error that stress is proportional to strain and that the direct modulus of elasticity in compression is equal to that in tension. So far then as these assumptions are concerned there is no more objection to applying the theory of curved bars to curved members of cast iron than to treating straight or slightly curved members of cast iron in the ordinary way.

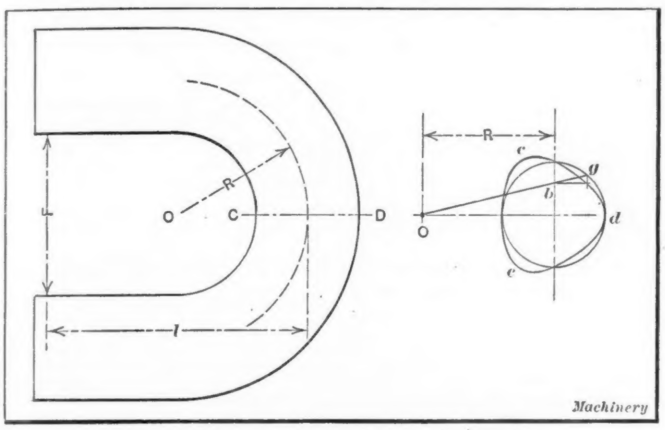


Fig. 1. Diagram indicating Notation in Formulas

The stress in pounds per square inch at any distance x from the gravity axis of the section CD, Fig. 1, is,

$$S = \frac{F}{A}\left[\frac{l}{R-x}\left(\frac{x}{R} \times \frac{a}{a-A} - 1\right) + 1\right] \qquad (1)‡$$

where x is positive if measured toward, and negative if measured away from, the center of curvature O.

F = the force or load in pounds acting at a distance of l inches from the gravity axis;

A = the area of the section CD in square inches;

R = the radius of curvature at section CD of the curve passing through the centers of gravity of the sections of the member;

$a = \int\left(\frac{R}{R-x}\right)\delta A$ = the area in square inches of a figure derived from the original section CD.

The expression given above takes account of the curvature of the member, but does not, as does the more complete analysis of Pearson and Andrews, take account of the lateral or transverse strains. For this reason it would, for the case illustrated in Fig. 1, indicate a somewhat lower maximum tensile stress and a somewhat higher maximum compressive stress than the more complete theory. To neglect, however, the lateral strains greatly simplifies the resulting expression, giving one that is quite easily handled and sufficiently accurate for all ordinary purposes, since in it account is taken of the curvature of the member which is the chief factor affecting the induced stress.

For irregular sections it is best to determine a graphically. Where, however, the section is made up of regular figures such as rectangles, triangles, or trapezoids, it is, as a rule, more accurate and easier to determine a algebraically. In any case a should be very carefully determined, since it is evident that any appreciable error will affect the ratio $\frac{a}{a-A}$, and therefore the results, quite seriously.

In Fig. 1, to the right, is shown how, from the original section, which in this case is circular, the derived figure cde is obtained graphically. From any point g on the outline of the section under consideration draw a radial line to the center of curvature O; through the point b where this radial line crosses the gravity axis draw a perpendicular; the intersection of this perpendicular through b with a line through g parallel to the gravity axis, locates a point on the outline of the derived figure cde the area of which is a.

Application to Punch Frame Sections

In Fig. 2 is shown a form of hollow section often used for punching machine frames, the filleting and rounding of the corners having been omitted. This section can for convenience be drawn in the form shown to the right and treated as made up of the rectangles A, B and D, and the trapezoid C.

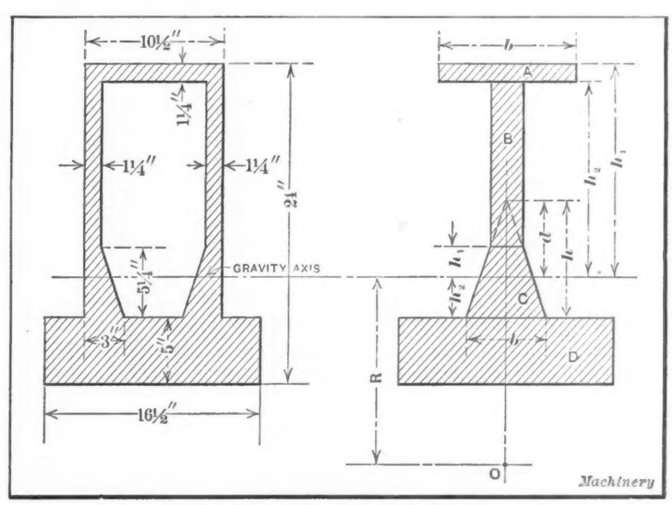


Fig. 2. Typical Section of Punch Frame

For any one of the rectangles A, B, or D:

$$a = \int\left(\frac{R}{R-x}\right)\delta A = Rb \text{ hyp. log}\left(\frac{R-h_1}{R-h_2}\right)$$

For the trapezoid C (or a triangle):

$$a = \int\left(\frac{R}{R-x}\right)\delta A =$$

$$R\frac{b}{h}\left[(d+R)\text{ hyp. log}\left(\frac{R-h_1}{R-h_2}\right) - h_2 + h_1\right]$$

To illustrate the purpose of this article as stated in the opening paragraph, a cast-iron frame for a machine to punch ⅝-inch holes in ⅝-inch plate will be taken up, the maximum resistance to punching being taken at 77,000 pounds. The frame and general dimensions shown in Fig. 3 will be found to be in fair agreement with machines of a like capacity.

In order that the radius of curvature for the horizontal or 0-degree section might in all cases be taken as the inner

* The following articles on kindred subjects have previously been published in MACHINERY: "The Design of Curved Machine Members under Eccentric Loads," December, 1909; "The Designing of Machine Frames," August, 1908; "Strength of Punch and Shear Frames," February, 1907; "Design of Punch and Riveter Frames," November, 1903.

† Assistant Professor, Department of Machine Design and Construction, Sibley College of Mechanical Engineering, Cornell University, Ithaca, N. Y.

‡ The expression given is the same as that in Morley's "Strength of Materials," page 325, although it is given here in a slightly different form.

radius of the throat plus the distance from the gravity axis to the outermost fiber in tension, the cross-section of the frames for 20 degrees each side of the horizontal was kept the same. Furthermore, for the frame outlines illustrated in Figs. 3, 4 and 5, the 0-, 20-, 45-, 67½- and 90-degree sections have been kept the same in each of the three designs shown, the changes in outline being due to changes in the inner radius of the throat. The horizontal section for these frames is shown in Fig. 2.

For the purpose of the conclusions to be drawn the maximum tensile stress at the horizontal section and at the 45-degree section for each of the three frames illustrated in Figs. 3, 4, and 5 will next be computed. First the curvature of the frames will be neglected and the stresses found in the ordinary way; then the curvature of the frames will be considered and the corresponding stresses found.

For the frame shown in Fig. 3 the following values obtain:

Section	0°	20°	45°	67.5°	90°
e	8	8	7.65	6.95	6.05
A	149.20	149.2	145.75	....	126.60
I	8487.00	....	7582.70	....	....
a	190.08	....	239.64	....	....
R	14.00	....	9.75	....	....

Machinery

In the table above,

e = distance in inches from gravity axis to the outermost fiber in tension;
A = area of section in square inches;
I = moment of inertia of section;
a = area in square inches of derived figure;
R = radius of curvature in inches.

Calculations Taking no Account of Curvature

Neglecting the curvature of the frames, the maximum tensile stress at the horizontal section for each of the three frames illustrated in Figs. 3, 4, and 5 is:

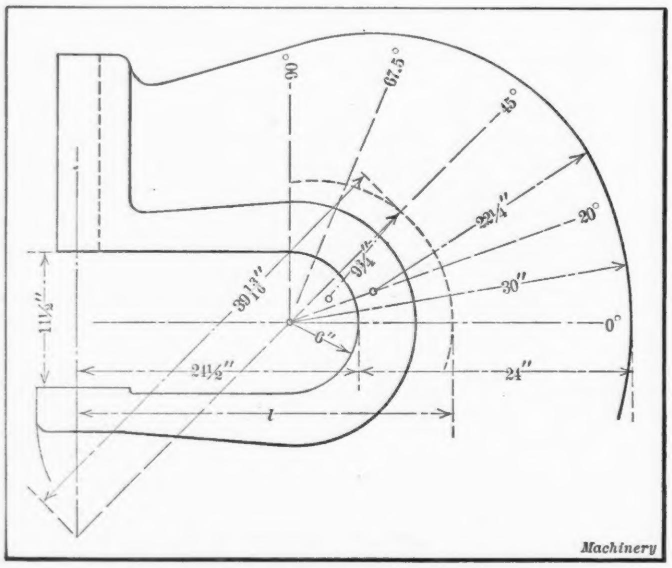


Fig. 3. Design of Punch Frame as commonly made

$$S_t = \frac{F}{A} + \frac{Fle}{I} = \frac{77{,}000}{149.2} + \frac{77{,}000 \times 32.5 \times 8}{8487} = 2876 \text{ pounds per square inch.}$$

The maximum tensile stress at the 45-degree section due to flexure and direct stress is, for the frame shown in Fig. 3:

$$S_t = \frac{F\cos 45}{A} + \frac{(F\cos 45)\,le}{I} = \frac{54{,}450}{145.75} + \frac{54{,}450 \times 39.8125 \times 7.65}{7582.7} = 2560 \text{ pounds per square inch.}$$

The shearing stress over this section is $S_s = (F \sin 45) \div A = 374$, and the maximum resultant tensile stress is:

$$S = \tfrac{1}{2}\left[S_t + \sqrt{S_t^2 + 4\,S_s^2}\right] = \tfrac{1}{2}\left[2560 + \sqrt{2560^2 + 4 \times (374)^2}\right]$$
$$= 2615 \text{ pounds per square inch.}$$

For the frame shown in Fig. 4, the distance from the center of gravity of the 45-degree section to the intersection of the 45-degree plane with the line of action of the load is 39 inches; for the frame shown in Fig. 5 this distance is 38⅛ inches. The 45-degree sections for these frames being the same as for the frame in Fig. 3, the maximum resultant

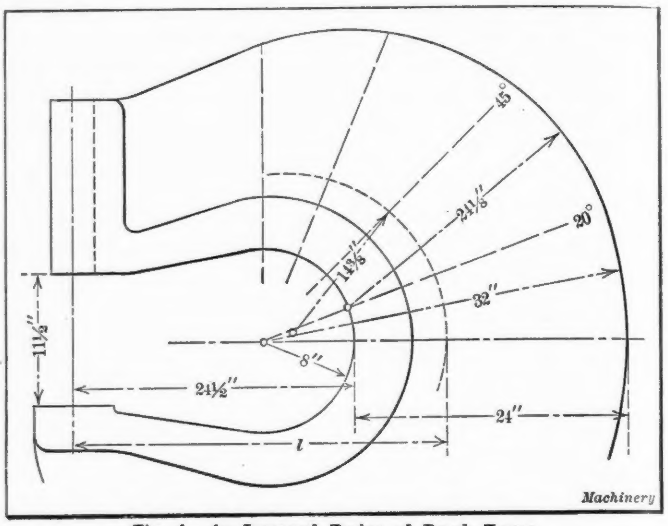


Fig. 4. An Improved Design of Punch Frame

tensile stress at the 45-degree section is for Fig. 4, 2566, and for Fig. 5, 2525 pounds per square inch.

Thus it is seen that, disregarding curvature, the frames are practically of uniform strength, but slightly weaker at the horizontal than at the 45-degree section.

Calculations Taking Account of Curvature

Considering the curvature of the frames the maximum tensile stress at the horizontal section for the frame in Fig. 3, is:

$$S_t = \frac{F}{A}\left[\frac{l}{R-e}\left(\frac{e}{R} \times \frac{a}{a-A} - 1\right) + 1\right]$$
$$= \frac{77{,}000}{149.2}\left[\frac{32.5}{14-8}\left(\frac{8}{14} \times \frac{190.08}{190.08-149.2} - 1\right) + 1\right] = 5150$$

pounds per square inch, or a value 1.79 times as great as that indicated by the ordinary method of computation.

For the 45-degree section frame, Fig. 3, the values obtained when considering curvature are more or less approximate, since the center of curvature does not fall in the plane of the section. So far, however, as the probable maximum stress in the section is concerned the calculation, considering curvature, is far more indicative of the true state of stress than the ordinary method not considering curvature.

The maximum tensile stress in the 45-degree section due to flexure and direct stress is for the frame in Fig. 3:

$$S_t = \frac{F\cos 45}{A}\left[\frac{l}{R-e}\left(\frac{e}{R} \times \frac{a}{a-A} - 1\right) + 1\right]$$
$$= \frac{54450}{145.75}\left[\frac{39.8125}{9.75-7.65}\left(\frac{7.65}{9.75} \times \frac{239.64}{239.64-145.75} - 1\right) + 1\right]$$

= 7450 pounds per square inch.

The augmentation due to the direct shear would raise this value, computed by the method illustrated above, to approximately 7475, or a value 2.86 times as great as that indicated by the ordinary method neglecting curvature.

The radius of curvature for the 45-degree section was determined graphically. While by this method it is perhaps impossible to say just what the exact value of the radius is, it is entirely possible to obtain a value slightly less rather than over the exact value, and thus be on the safe side.

For the frames in Figs. 4 and 5, the inner radius of the throat has been changed from 6 inches to 8 and 10 inches, respectively. The radius of curvature at each of the various sections is therefore increased with a consequent change in the value of a. For these frames the following values obtain:

	Horizontal Section		Forty-five Degree Section	
	Fig. 4	Fig. 5	Fig. 4	Fig. 5
e	8	8	7.65	7.65
A	149.2	149.2	145.75	145.75
a	179.14	172.64	178.135	173.45
R	16	18	14.375	15.625
l	32.5	32.5	39	38.125
				Machinery

Results Obtained from Calculations

Having the above values, the maximum tensile stresses at the horizontal and 45-degree sections can be found. Tabulating these and the values already found we have:

Conditions		Fig. 3	Fig. 4	Fig. 5
Horizontal section	Curvature neglected	2875	2875	2875
	Curvature considered	5150	4685	4320
45-degree section	Curvature neglected	2615	2566	2525
	Curvature considered	7475	4570	4110
				Machinery

So far as the horizontal section is concerned, the frame in Fig. 4 has an advantage of about 9 per cent, and the frame in Fig. 5 of about 16 per cent over the frame in Fig. 3. Now, since the breaking load for a frame cannot be com-

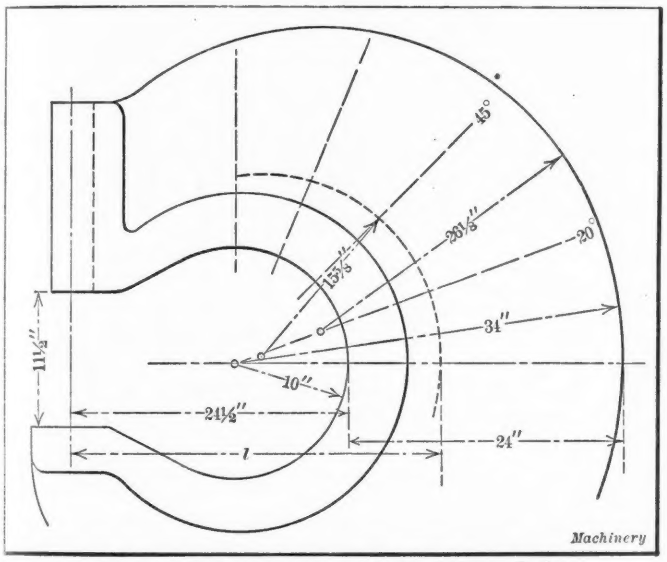


Fig. 5. A Further Improvement in Punch Frame Design

puted by either method, it would seem, for the particular frame taken up, that, in so far as the horizontal section is concerned, the curvature of the frame might be neglected in our computations, if a proper limiting stress be chosen. To make such a deduction on the basis of the above problem and to extend it to any punch frame would, however, be a mistake since it is evident from Formula (1) that the maximum induced stress in such a member depends very largely on the relation of the radius of curvature to the size and general proportions of the section. The results of the computations on the 45-degree sections very well illustrate this.

The computations, curvature considered, show the frame in Fig. 3 to be considerably weaker at the 45-degree section than at the horizontal section; the computations neglecting curvature show the reverse of this. The tabulation, curvature considered, shows the stress in the 45-degree section for the frame in Fig. 3 to be much higher than that in the horizontal section, and the stresses in the horizontal sections of the frames in Figs. 4 and 5 to be higher than in the 45-degree sections. On this basis the frame in Fig. 4 has an advantage of approximately 37.5 per cent and the frame in Fig. 5 an advantage of approximately 42 per cent over that in Fig. 3.

Conclusion

Apparently, then, that there may not be any abrupt changes of stress in the frame, there should not be any abrupt changes of curvature in the curve passing through the centers of gravity of the sections. Furthermore, the radius of curvature at the horizontal section should be as great as possible and the value of the radius of curvature at any other section as we pass forward should be as little below that at the horizontal as possible. The inner outline of the throat should, therefore, be a smooth curve with no abrupt changes of curvature. The frames in Figs. 4 and 5 suggest a way in which the above requirements may be met, due account being taken of appearance and the general limitations of the problem, and are much to be preferred to the frame in Fig. 3.

* * *

THE SAFEGUARDING OF BELTS, SHAFTS AND PULLEYS

The following concise rules have been issued by a large British steel company with a view to preventing accidents in the company's mills, arising from moving belts, shafts or pulleys. The rules are practical and concisely stated, and may be of considerable value in formulating similar rules to meet the conditions in other plants:

Belt, rope, and cable transmission should be arranged, so far as possible, to be easily guarded.

Pipe or angle railing placed 18 inches from the pulley or belt should be provided where there is but little danger of persons getting caught by the belt.

Wire-mesh or perforated-plate casing should be provided where the location of the belt is such that persons are likely to get caught between the belt and the pulley.

Where the belt is located in such a position that, should it break, flying ends would strike passers-by, it should be enclosed by a suitable guard.

The width of a belt should be allowed between two pulleys or a pulley and a hanger on shafting so as to prevent the belt from becoming wedged and possibly pulling the line-shafting down.

Belt splicing should be made so as to eliminate ragged edges or projections which might catch employes' clothing. Endless belts are recommended, especially for high-speed machines.

Belts should be inspected frequently and kept in a condition to avoid accidents. Tight belts should be avoided.

Wherever it is necessary to shift belts a mechanical device should be provided.

Where it is necessary to unship a belt, a hook or belt-perch should be provided to prevent the belt from resting on the shaft.

Shafting not over 12 inches above floor should be covered.

Shafting over 12 inches and not over 7 feet above floor should be encased or railed off.

Exposed ends of shafting should be encased or otherwise guarded.

Overhead shafting which must be oiled should be equipped with a walk for the oiler, if practicable.

Electrically-driven shafting should be provided with a safety switch, to be used when men are working on it. This switch should be placed at the top of the ladder or in some place where no one will tamper with it.

Shaft couplings should be guarded where a safety coupling is not used.

To prevent gears or pulleys from working off the ends of shafting, shaft ends should project at least the diameter of the shaft beyond the hub of the gear or pulley, so that the key can be locked in place.

Hangers for shafting should be particularly strong and well secured.

All projecting keys in shafting, when exposed so that a person might get caught thereon, should be guarded or cut off.

Wherever possible, cotter pins or spring washers should be provided to prevent nuts from working off the ends of bolts.

Set-screws should be guarded, countersunk, or placed in safety collars with flanges high enough so that the set-screw head will not project above the flange.

* * *

The man who works with one eye on the clock and the other on the boss, and who is the first one out of the door at night, is provided with an automatic resignation.

POWER REQUIRED FOR ROLLING METAL*

Twenty-eight years ago the writer was asked to compute the power required to roll billets to rods, and also the power required to draw the same rods to wire. It was first necessary to assume a common unit for comparison. One base only appeared rational, the power required to cause a unit reduction of a unit mass in a unit time. The units the writer first adopted were a unit reduction of 50 per cent or doubling of the length; a unit mass of one ton; and a time unit of one minute. This gave a formula

$$H.\ P. = CTD$$

where C = constant;

T = tons worked per minute;

D = doublings or number of times the length has been doubled.

A table of lengths and doublings was prepared corresponding with each successive per cent reduction from 1 to 99 per cent. The last column of this table was a table of logarithms having 2 for a base. For example:

50 per cent reduction = length 2.00 log 2.00 = 1;

75 per cent reduction = length 4.00 log 4.00 = 2; etc.

As is well known, the logarithms of any base are made by multiplying common logarithms by the reciprocal of the common logarithm of the desired base, or, in other words, by a constant. Hence, the formula is more available when common logarithms are used, and it becomes

$$H.\ P. = CT \log E$$

where

$$E = \text{elongation} = \frac{\text{length after rolling}}{\text{length before rolling}}.$$

This simply changes our unit reduction from 50 to 90 per cent; 90 per cent reduction gives an elongation of 10 to 1, and 10 is the base of our common logarithms which are universally available. In the above formula, C is a constant when other conditions are constant, but other conditions are seldom constant. Careful and extended dynamometer tests of cold-rolling relatively wide strips of commercially pure aluminum showed clearly that the power required increased somewhat faster than $\sqrt{\frac{D}{t}}$ where

D = roll diameter,

t = finished thickness.

This changes the formula to

$$H.\ P. = CT \log E \sqrt{\frac{D}{t}}.$$

Commercially pure aluminum was chosen because its malleability is perhaps exceeded only by gold and silver, and it does not harden appreciably with continued rolling. Averages of repeated tests plotted on logarithmic cross-section paper from the above formula appear to indicate that the power does increase almost directly as $\sqrt{\frac{D}{t}}$, between the values $\sqrt{\frac{D}{t}} = 5$ and $\sqrt{\frac{D}{t}} = 10$. After passing the latter point, the power required increases more rapidly, until for values $\sqrt{\frac{D}{t}} = 15$ the unit power required was 30 to 50 per cent more than that called for by the formula. Perhaps this was partly due to the hardening of the aluminum by the continued rolling, but is not much of it due to the increased ratio between the arc of contact and the thickness of the metal? In other words, is it not true that more pressure per unit area is required to compress a very thin sheet of metal between two flat plates than would be required to compress a much thicker sheet? The diagram previously referred to, for showing the relative work required to roll plates of varying final thickness and percentage reductions, also seemed to indicate that from 40 to 60 per cent reductions are the most economical in the use of power; that reductions less than 30 per cent are extravagant in power consumption, due perhaps to an excessive proportion of journal friction; and that rough rolls require much more power than smooth rolls.

Temperature is another factor nearly as important as roll diameter. Someone has covered this in what appears to be a perfectly logical way, by assuming that the power varies directly with the tensile strength of the bar being rolled. Adding this makes the formula quite complete:

$$H.\ P = CT \log E \sqrt{\frac{D}{t}}\, S$$

in which S = tensile strength at the rolling temperature.

This formula clearly shows why a continuous sheet bar mill may require more power to deliver a bar ¼ inch thick than it does to deliver twice the tonnage at the same speed of a bar ½ inch thick, and only slightly warmer. It is interesting and easy to remember that through nearly the entire range of rolling temperatures, a reduction of 400 degrees F. practically doubles the tensile strength and hence the power required to roll. The formula also shows why small rolls and high speeds will conserve temperature and power.

At least one more factor is necessary to make the formula complete, *viz.*, the efficiency of the pass. Obviously, the metal can take the speed of the roll at one point only in the arc of contact. At all other points the metal must slip either against the face of the roll or internally. As an example, it is well known that a barrel hoop may deliver fully 10 per cent faster than the surface speed of the roll. This slipping, either external or internal, requires and wastes power. The smoother the roll, the less power wasted.

Cold-rolling is done in oil and all rolls for cold work are ground as true and smooth as possible. Should not the same reason lead us to grind with much care all rolls for hot work that can be so finished? Less power is required, the rolls last longer and better work is produced. The foregoing applies to plain, flat passes.

The efficiency of all shaped passes decreases as the slipping increases. This loss of efficiency is especially noted in some of the passes used for rolling rails, but the benefits from using such passes may fully warrant the resulting excess roll wear and power. Light reductions have a low efficiency. It may not be easy to get a formula that will cover the efficiency of a pass, but it should not be overlooked in computing power requirements.

* Abstract of a paper presented before the Engineers' Society of Western Pennsylvania, Pittsburg, Pa., by Victor E. Edwards.

* * *

UNIQUE POWER PLANT FOR FORD MOTOR CO

The Ford Motor Co. has placed contracts for a gas engine-electric power plant that will be not only one of the largest in the country, but in many respects absolutely unique. The company has appropriated about a million dollars for the project, which will put into effect plans that Henry Ford has long had in mind for utilizing the waste heat of the ordinary producer gas engine. Four 6000 H. P. Hamilton-Gray gas engines of novel design will drive four Crocker-Wheeler 3750 K.W., 250 volt, 80 R. P. M. engine type direct-current generators. A plan view of each engine will be similar to a cross-compound steam engine, with two cylinders in tandem on each side. One pair of cylinders will be operated by producer gas and the other by steam. The steam will be generated from the water used in the water jacket of the gas engine, further heated by the exhaust gases and by waste heat from the producer gas plant. This water or steam will be used as the feed water for the boiler which supplies the steam engine cylinders. A heavy flywheel will equalize the characteristics of the gas and steam driven elements of the engine. These generators are designed for much higher efficiency than is ordinarily found in commercial practice. Full load efficiencies will be not less than 94½ per cent. By these means and by the utilization of energy usually lost in waste heat, it is proposed to make the new Ford power plant the most economical in the country in respect to cost of production per kilowatt hour. The armatures will be of split construction which is necessitated by clearance requirements through tunnels and bridges in shipment from Ampere to Detroit, and the generators will be finally assembled at the Ford plant.

MAKING SECTIONAL DIE PARTS*

SOME TOOL-MAKING METHODS AND TOOLS EMPLOYED BY THE COLUMBUS DIE, TOOL & MACHINE CO.

BY DOUGLAS T. HAMILTON†

The making of sectional dies for armature disks and work of a similar kind calls for considerable ingenuity on the part of the toolmaker in devising means for producing these parts accurately and economically. It is necessary, if the best results are to be obtained, to secure a steel which will not warp or distort during the hardening process so that grinding of the parts after hardening need not be resorted to. Should the die and punch sections warp or shrink, it adds greatly to the difficulty of manufacturing the parts and greatly increases the cost of the tool. In many cases grinding of the die and punch sections is necessary, so that special devices and tools have been devised for handling this work as expeditiously as possible. In the following article, a number of the most interesting methods employed by the Columbus Die, Tool & Machine Co. of Columbus, Ohio, will be given. This

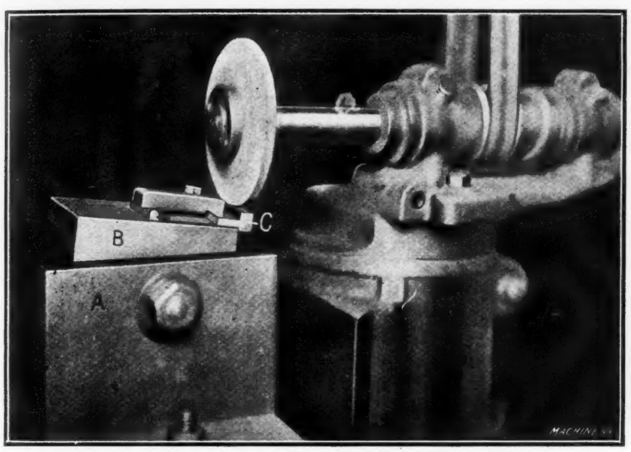


Fig. 1. Grinding the Die Sections

company makes a specialty of producing for various firms punches, dies, tools and special equipment, and therefore is in a position to devise means for getting this work out in the shortest possible time.

Grinding the Die Sections

In order that the die sections will fit properly in place after hardening and give the required outside and inside diameters, it is necessary to leave a slight amount of excess stock on the sides of the sections that fit against each other in the die holder. These sides are then ground to the required angle and thickness in a special fixture. The toolmaker figures out the required angle of the side sections and also the least or greatest thickness. The type of fixture used for grinding

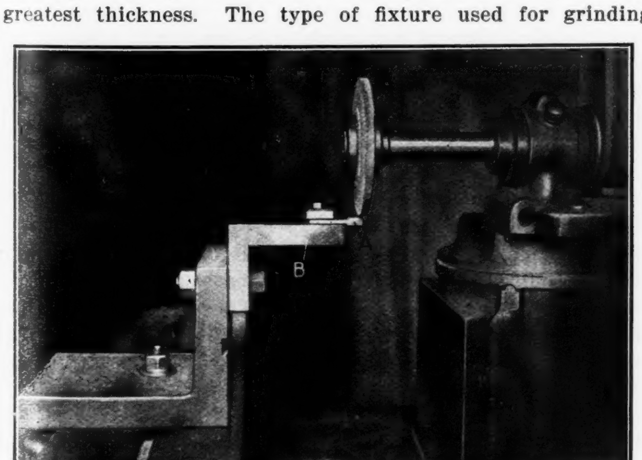


Fig. 2. Grinding the Punch Sections

these die sections is shown in Fig. 1. It consists of an ordinary angle-plate *A* which is fastened to the table of a grinder of the surface type, and against which is held another angle-plate *B* that can be set around to the desired angle and carries the work to be ground. The work or die section *C* in this case is held in place by a special toe clamp, and when one side of the section is ground it is reversed and the other side is ground.

The sides of the punch sections are ground in a similar

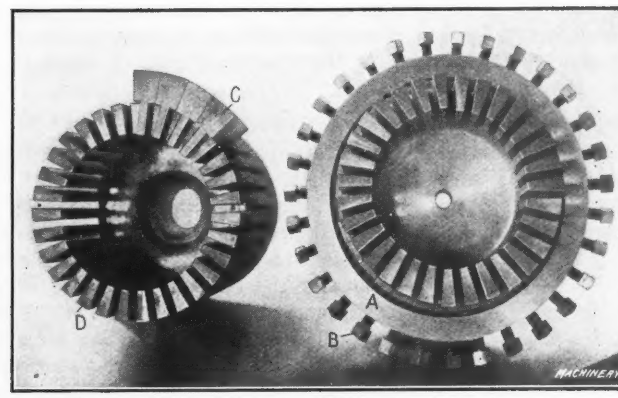


Fig. 3. Special Chucks for holding Die and Punch Sections while Grinding

manner on the same fixture which, however, is swung around into the position shown in Fig. 2. The punch section *A* is held to the small angle-plate *B* as illustrated, and when one side is ground the section is reversed on the angle-plate and the other side finished. When the fixture has once been set up, however, all the punch or die sections are ground on one side first. Then the setting of the machine is changed and the other side of all the sections ground. This enables the toolmaker to turn out the work much more quickly than if

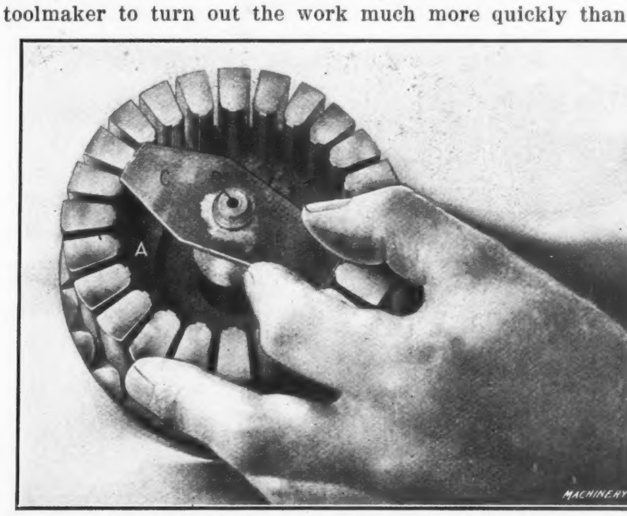


Fig. 4. Assembling a Sectional Armature Disk Punch

he were to reset the machine for both sides of the piece, thus finishing it complete without removing it from the fixture to put another in its place.

Special Chucks for Holding Die and Punch Sections while Grinding

In Fig. 3 are shown two special chucks which are used for holding punch and die sections while grinding the inside and outside diameters and the top and bottom faces, so that all the important machined surfaces can be finished at the same setting. The special chuck for holding the die section is shown to the right of the illustration. This, as can be clearly seen, consists of a cup-shaped body *A* around the periphery of which are located set-screws *B*. These set-screws bear against the backs of the section and bind them together, the beveled surfaces of the punch section being wedged together by the action of the set-screws, and consequently held rigidly in place for the grinding operation. The grinding is accomplished in a cylindrical grinding machine, the chuck being screwed to the nose of the spindle in the ordinary manner. The outside surfaces are ground tapered to an angle of 5 degrees to enable them to be held in place in the punch holder by a retaining ring.

* For further information on sectional punch and die work, see "Punch and Die made in Sections," March, 1913, and "Sectional Punch and Die Construction," July, 1913.

†Associate Editor of MACHINERY.

The special chuck for holding the die sections for grinding is shown to the left of the illustration. Here only four die sections *C* are shown in place, simply to indicate the manner in which they are held. The narrow portion of the die section is a drive fit in the slots of the die holder and these sections are tapped lightly into place until the beveled surfaces contact. The chuck *D* is also held in a cylindrical grinder, being screwed to the nose of the spindle as previously mentioned in connection with the chuck for the punches. This chuck enables the inside and outside diameters of the die sections to be ground and also the top face. The lower face is ground by reversing the position of these sections in the chuck, and then holding them in the manner in which they are held in the illustration.

Assembling an Armature Disk Punch

After the beveled sides of the punch sections have been ground, it is then necessary to test these sections to see whether the correct inside and outside diameters have been secured, and also if the small projections on the inner surfaces of the punch sections are properly located axially. This, of course, proves whether the correct amount of stock has been ground from the sides of the punch sections. The fixture used for this purpose is shown in Fig. 4 and consists of a block *A*, circular in shape, in which a stud *B* is located. This stud acts as a means for holding the gage *C*, which as can be seen, is cut out to fit over the projections on the punch sections and is also fitted over the stud, being centrally located in this manner. In this particular case, a limit of only 0.0005 inch is allowed from the center of the plug to the inside face of the punch section. Also the blank must be reversible on the punch. When it is realized that these dimensions are governed entirely by the amount of metal removed from the sides of the sections, it will be seen that the grinding operation is one that must be very carefully handled and that requires considerable ingenuity on the part of the toolmaker if the parts are to assemble accurately.

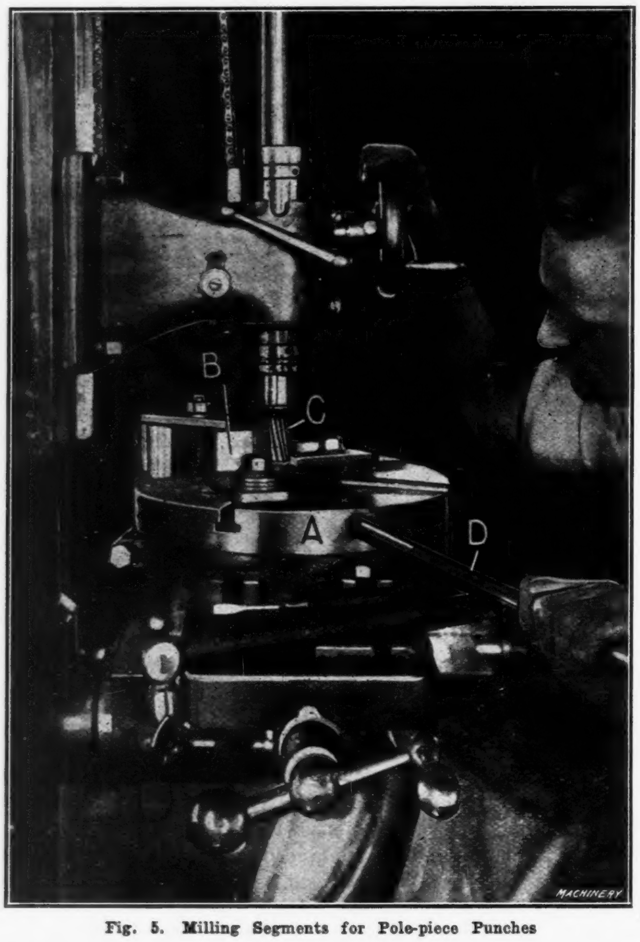


Fig. 5. Milling Segments for Pole-piece Punches

Milling Segments for Pole-piece Sectional Punches

Fig. 5 shows how segments for pole-piece sectional punches are milled both on their circular and angular faces. A Knight milling and drilling machine is used for this purpose, being equipped with a circular milling attachment *A*. The section *B* of the pole-piece punch to be machined is clamped to the top face of the circular milling attachment by clamps as illustrated, and the machining is accomplished by an end-mill *C* held in the spindle of the machine. The circular attachment is operated by a bar *D* which can be located in holes provided around the periphery of the faceplate. For milling the beveled faces, the circular attachment is clamped to prevent it from rotating and the feed-screw for the table is operated.

Boring Sub-press Die Guide Pin Holes

After all the various members of the sectional punch and die have been completed, they are assembled in the punch and die holders and then the next operation is to bore the holes for the aligning pins, when the dies are of the sub-press construction, which is the usual type of construction adopted for complicated sectional punches and dies. The manner in which these guide pin holes are machined is shown in Fig. 6. The punch and die holder are located in the proper relation to each other and are fastened together by through bolts as illustrated. Then these holders for the sectional members are located on the table of a milling machine and the holes are first drilled and then bored by a boring tool held in the spindle of the machine. The two lower holes are bored and counterbored first, the center distances, of course, being located by the micrometer dial on the feed-screw. Then the table is lowered and the top or two remaining holes are bored and counterbored in a similar manner. This method of boring the guide pin holes enables them to be produced quickly and accurately with very little difficulty.

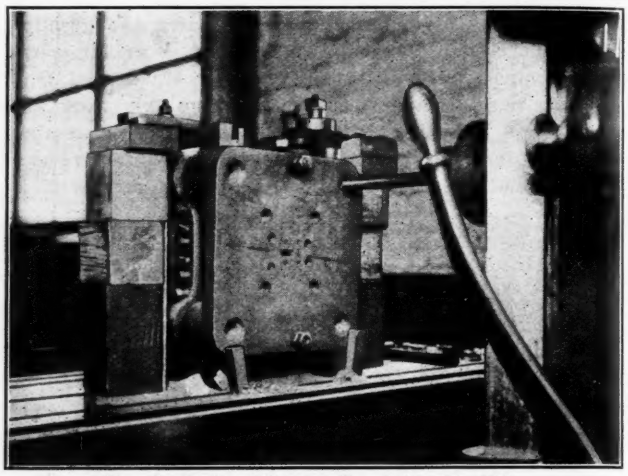


Fig. 6. Boring Sub-press Die Guide Pin Holes

* * *

One of the most important iron mines in the world is the Loussavaara-Kürunavaara, situated at Kiruna, Lapland, in latitude 68½ degrees north, which is about the same as the northernmost boundary of Alaska. The climate is somewhat milder than in Alaska, and these mines are worked the year round. About 1600 men are employed. The plant is operated by steam power, the coal coming from England and Spitzbergen. The electrification of the mine is in progress, and this power will be used exclusively as soon as the new power plant of the Swedish government, now under construction at Porjus Falls, is completed. The capacity of this station will be 150,000 H. P. to be transmitted at 78,000 volts over a distance of 150 miles. The ore is shipped from Kiruna to Narvik on the Norwegian coast by rail, and from there by water to Germany, a small percentage finding its way to other countries. The ore is magnetite and contains from 53 to 68½ per cent iron. Owing to the comparatively large content of phosphorus it was impossible to utilize it until the Thomas furnace was developed.—*Daily Consular Report.*

* * *

At the International and Colonial Exhibition to be held in Genoa, Italy, beginning in March, there will be an interesting section devoted to showing methods of packing for export. There will be nine different lines of goods exhibited with the proper packing for each, one of these lines being packing required for machinery.

DRILLING AND TAPPING FIXTURE

BY CHRISTIAN F. MEYER*

The drilling and tapping fixture to be described in this article was designed for machining holes in the bed of a textile machine. Fig. 1 shows a section of the bed *A*; this bed is about 30 feet long and is supported by legs *B*. After the bed has been machined, a number of other parts are assembled on it, the finished surface of the bed being used as a base in lining up the other members. When the assembling has been completed, a number of holes *a* are drilled and tapped in the bed to receive guide pins *C*. These guide pins are screwed into different holes in the bed according to the kind of fabric that is to be woven, and it is very important to have them exactly perpendicular to the top surface of the bed *A*. They must also be accurately located in relation to other members of the machine. These conditions make it impossible to remove the bed from the machine, after it has been assembled, in order to drill the holes, and as a result it was necessary to devise a fixture which would make it feasible for the holes to be drilled and tapped with the bed in position. The drilling was formerly done with an electric drill, after which the holes were tapped by hand. This method required a considerable amount of time and the results obtained were unsatisfactory. Great care and a good deal of skill had to be exercised in

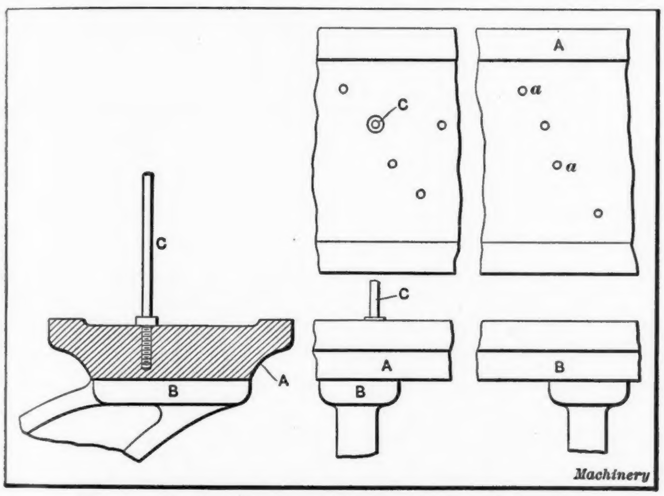


Fig. 1. Machine Bed for which the Drilling and Tapping Fixture was designed

order to get the holes perpendicular to the upper surface of the bed.

In order to eliminate the difficulties experienced in machining these holes by hand, the drilling and tapping fixture illustrated in Figs. 2, 3 and 4 was designed. It consists of a cast-iron base *D* which is made to slide along the surface of the bed *A*. After the longitudinal position of the fixture has been adjusted the fixture is held securely by means of hexagonal headed set-screws and a clamping gib. It will be seen that this gib is held in place in the fixture by means of the cover *E*. The base *D* supports an upper slide *F* which is held in any desired position by means of a clamping gib and set-screws. Referring to the plan and sectional views shown in Figs. 3 and 4, it will be seen that the slide *F* has a lug *G* at its rear end. This lug is tapped to receive an adjusting screw which is held against longitudinal movement by means of collars at each side of the end plate *H*.

It will be seen that the slide *F* carries a round boss about which the bushing *I* can be revolved. This bushing is held in any desired position by means of a set-screw. The tool-holder *K* revolves about the bushing *I* and is supported by the flange at its lower end. It will be seen from Figs. 2 and 3 that the tool-holder is provided with three arms *L*, *M* and *N*. The centering arm *L* is provided with a long "feeler" which slides in a vertical bushing and can be fastened in the desired position by a set-screw. The drill-arm *M* carries a steel bushing which is provided with rack-teeth. The bushing and rack slide on the arm, the movement being obtained by a pinion which provides for feeding the drill. The pinion shaft has a hub at its outer end which is provided with a hand-lever *T* for operating the feed mechanism. The drill spindle *W* revolves in the bushing and is driven by a direct-connected motor *R* which is mounted on a bracket at the rear of the tool-holder. It will be seen that this motor transmits through a pair of spur gears, the shaft *k* and the bevel gears *Q*. The bracket *S* is provided at the front of the tool-holder to support the shaft *k* and the lower bevel gear *Q*. The movement of the drill spindle is limited by a stop-pin *m* carried by the bushing which slides in a slot in the arm *M*. The weight of the spindle is counterbalanced by a weight located inside the boss on the slide *F*.

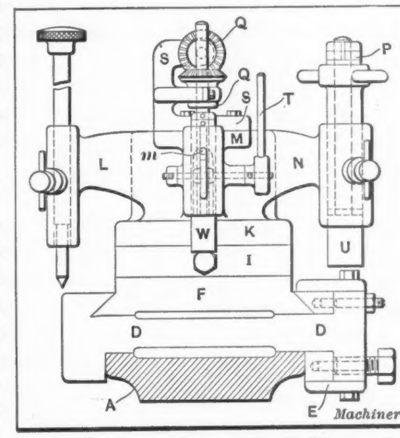


Fig. 2. Front View of the Drilling and Tapping Fixture

The tapping-arm *N* carries a steel bushing which may be held in any desired position by a set-screw. The spindle *U*, which revolves in this bushing is made with its lower end of the same diameter as the bushing. This section of the spindle is separated from the bushing by means of a leather washer, and two nuts *P* are used to draw the spindle *U* up against the base of the bushing. It will be seen that a similar leather washer is provided between the lower nut *P* and the top of the bushing. The bushing is turned by hand levers and when the nuts *P* are tightened, it will be evident that the spindle is driven by the friction of the leather washer. If the resistance of the tap becomes great enough to overcome this friction, the rotation of the spindle will be stopped, and the friction is adjusted so that slippage will occur before the resistance is great enough to break the tap.

It will be seen from the sectional view, Fig. 4, that the tool-holder *K* carries a boss on the opposite side to the arm *M*. This boss is drilled to receive the taper index pin *V* which is pressed inward by a compression spring. The taper section of the pin *V* fits into three corresponding holes in the bushing

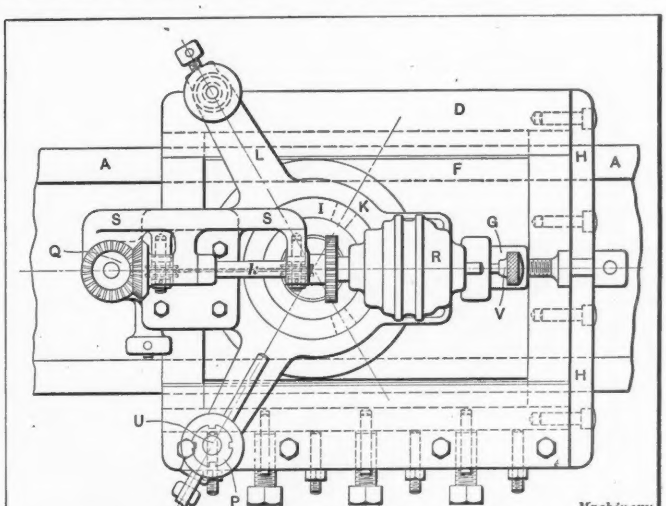


Fig. 3. Plan View of the Drilling and Tapping Fixture

I, these holes being drilled radially so that their center lines and the center lines of the arms *L*, *M* and *N* all intersect at the center of the bushing. The index holes in the bushing are located 60 degrees from each other, the arms of the tool-holder being located similar distances apart. From the preceding it will be evident that the three spindles of the fixture will be located at exactly the same point over the work when held by inserting the index pin into successive holes in the bushing *I*.

The operation of this drilling and tapping fixture may be briefly described as follows: The holes to be drilled and tapped are first laid out on the top surface of the bed of the machine. The fixture is then mounted in position and slid along the bed to the first group of holes. It is fastened in this posi-

* Address: Garfield Ave., Wyomissing, Pa.

tion by tightening the set-screws against the gib. The feeler-arm L is next located by first inserting the index pin V in the proper hole; then by revolving the bushing I and using the adjusting screw to regulate the position of the slide F, the point of the center punch is brought into coincidence with the center of one of the holes. Bushing I is then fastened by tightening the set-screw. The index pin is then withdrawn and the tool-holder K is revolved through 60 degrees, when the index pin is forced into the next hole in the bushing I by the action of the compression spring. This brings the drill spindle into the operating position so that the hole can be drilled. As the depth of the hole is governed by the stop-pin m, the operator does not have to waste any time in gaging. After the hole has been drilled, the fixture is again indexed to bring the tapping spindle into the operating position, after which the hole is tapped.

The use of this fixture insures having the drill and tap held exactly perpendicular to the surface of the work so that an inexperienced workman can machine the holes more quickly and accurately than a skilled mechanic could when the old hand-method was used. Although this drilling and tapping fixture was designed for the particular class of work referred

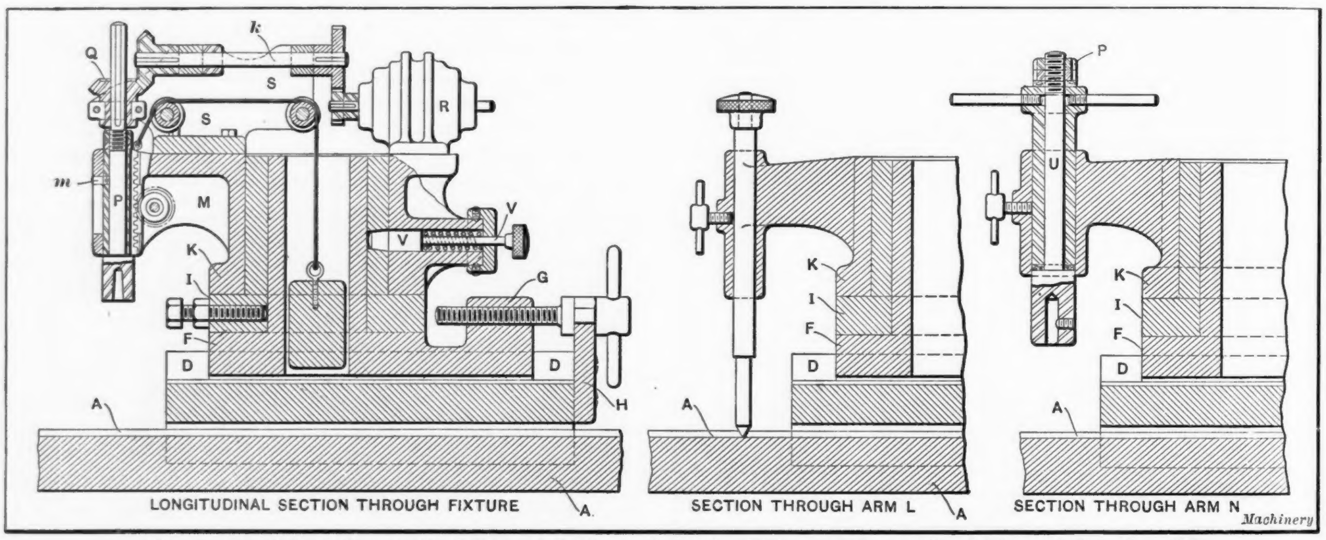


Fig. 4. Longitudinal Cross-sectional View of Drilling and Tapping Fixture and Cross-sectional Views of Arms L and N

to, the same device could be used to advantage on a variety of other classes of work.

* * *

As a rule, hard water causes little or no corrosion in metal pipes, principally because of the formation of protective coatings upon the metal by ingredients in the water. The carbonic acid and atmospheric oxygen in soft waters cause corrosion. Investigations made by the Department of Water Supply, New York City, indicate that the addition of lime or soda to soft water forms coatings on the inside of the pipes so that corrosion may be practically negligible.

BROACHING ROUND HOLES

BY RALPH R. LAPOINTE*

When finishing a hole by reaming, the cutting edge of each reamer blade travels a distance approximately equal to the

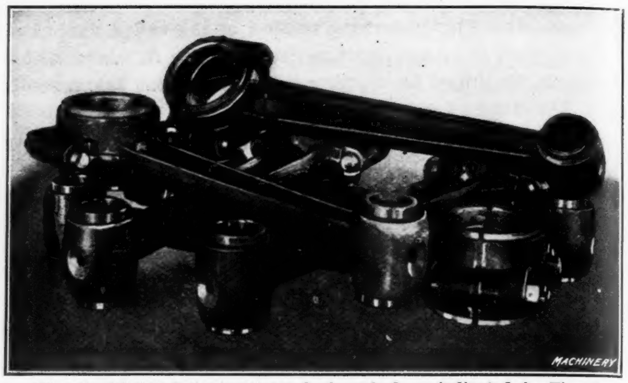


Fig. 1. Finished Connecting-rods broached as indicated in Fig. 2

circumference of the hole multiplied by the number of revolutions or turns made by the reamer in passing through the work. In the case of a round broach, the distance traversed by any point on the broach in machining a hole is equal to the length of the work; hence, it can easily be seen that broaching is a much faster operation than reaming, especially when the broach has a cutting speed of at least four feet per minute; moreover, the broach will maintain its size for a

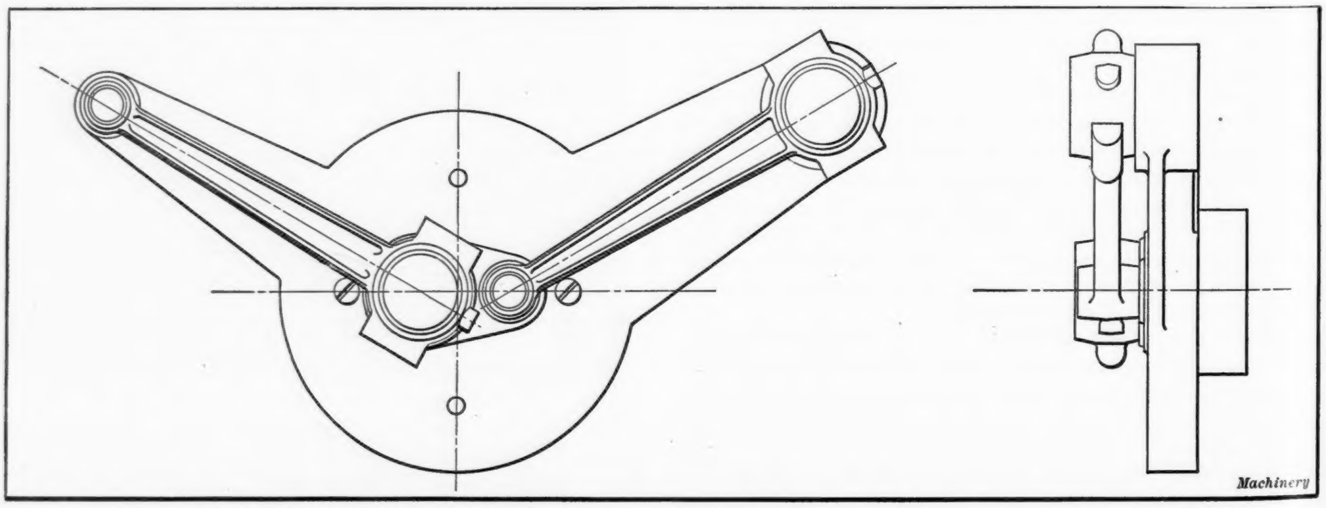


Fig. 2. Method of broaching Large and Small Connecting-rod Ends simultaneously on a Regular No. 3 Lapointe Broaching Machine

longer period than a solid reamer, because the finishing end of the broach has a number of teeth of the same diameter, and as these only take very light finishing cuts they are subjected to very little wear. Even an adjustable reamer has no longer life than a well made round broach.

* Superintendent, Lapointe Machine Tool Co., Hudson, Mass.

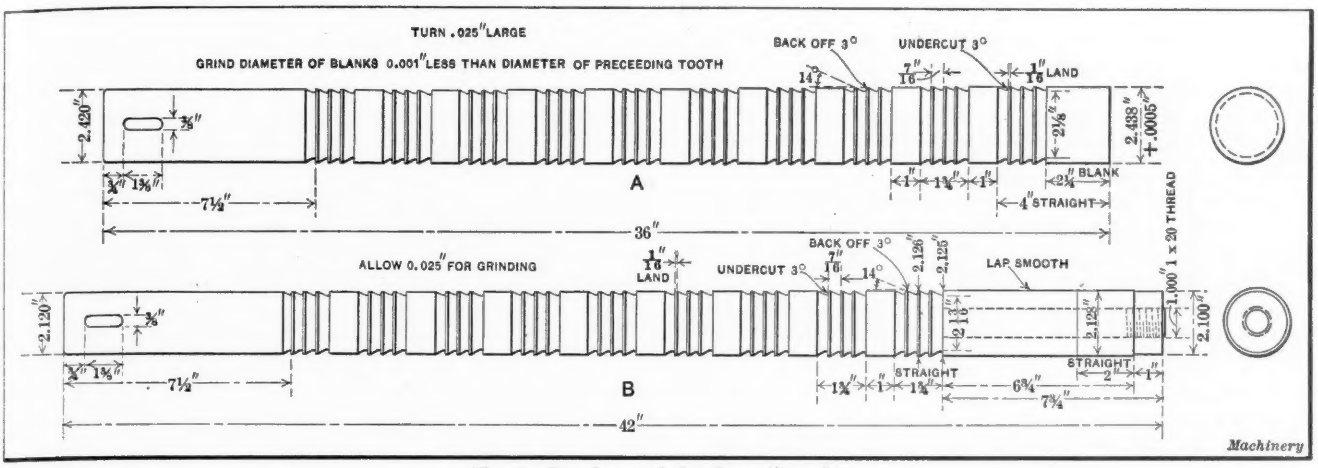


Fig. 3. Broaches used for Connecting-rods

It has been demonstrated that the broaching of hard chrome-nickel steel, such as is used in automobile work, is a much cheaper process than reaming. A typical job is shown in Fig. 2, which illustrates two connecting-rods and their

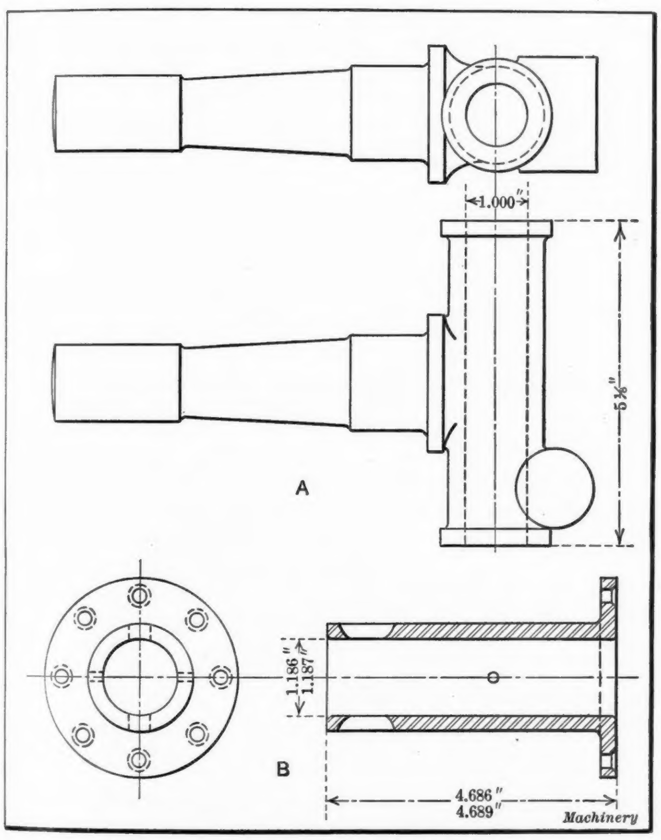


Fig. 4. Examples of Work machined by broaching

broaching fixture. The small end of one rod and the large end of the other are broached simultaneously on a regular No. 3 Lapointe broaching machine. In this way, one complete rod is finished for every stroke of the machine. The fixture is not absolutely necessary but adds considerably to the production. These connecting-rods are first drilled to the size of the broach shank or to a diameter of from 0.015 to 0.018 inch under the required size. They are then finished by broaching, thus eliminating both machine and hand reaming. After the rods have been broached, the large end is split and the lining bushing for the large end is inserted. The bushing for the small end is pressed into the rod. These bearings or bushings are then broached, and the finish obtained can only be duplicated by scraping, although broaching is, of course, much cheaper, and the metal is compressed somewhat, thus giving the bearing a hard, glazed surface that resists wear. A number of finished connecting-rods are shown in Fig. 1.

The broach illustrated at *A* in Fig. 3 is used for broaching the hole in the large end of the rod, whereas the smaller broach *B* is for finishing the bushing. The plain round sections seen on these broaches are for the purpose of keeping the broach from "running" or "crawling," as it is essential that the center-to-center distance of these rods be kept fairly accurate. By introducing plain blanks or sections between the teeth, as shown, the broach is kept properly aligned with the hole because there is always some portion of the blank section in the work while some of the teeth are cutting.

When using a broach, it is passed through the work and is fastened to the draw-head of the machine by a cotter-key which passes through the slotted end in the usual way. Only such clamping as is necessary to support the work is required, as the blank sections on the broach will hold the part in alignment. When using these broaches in cast iron, a soap cutting compound is used, as this gives the broached surface a highly polished finish. For chrome-nickel steel, a good grade of cutting oil will give satisfactory results. On some work, no drilling whatever is done prior to broaching, and very often only one broach is used, but if the work is longer than, say, two inches, a roughing broach usually precedes the finishing broach. Of course, broaching from the rough can only be done when the broaching operation comes first, as otherwise the broach would follow the rough hole and, consequently, the finished hole would be out of true with any other surfaces which might be machined afterward.

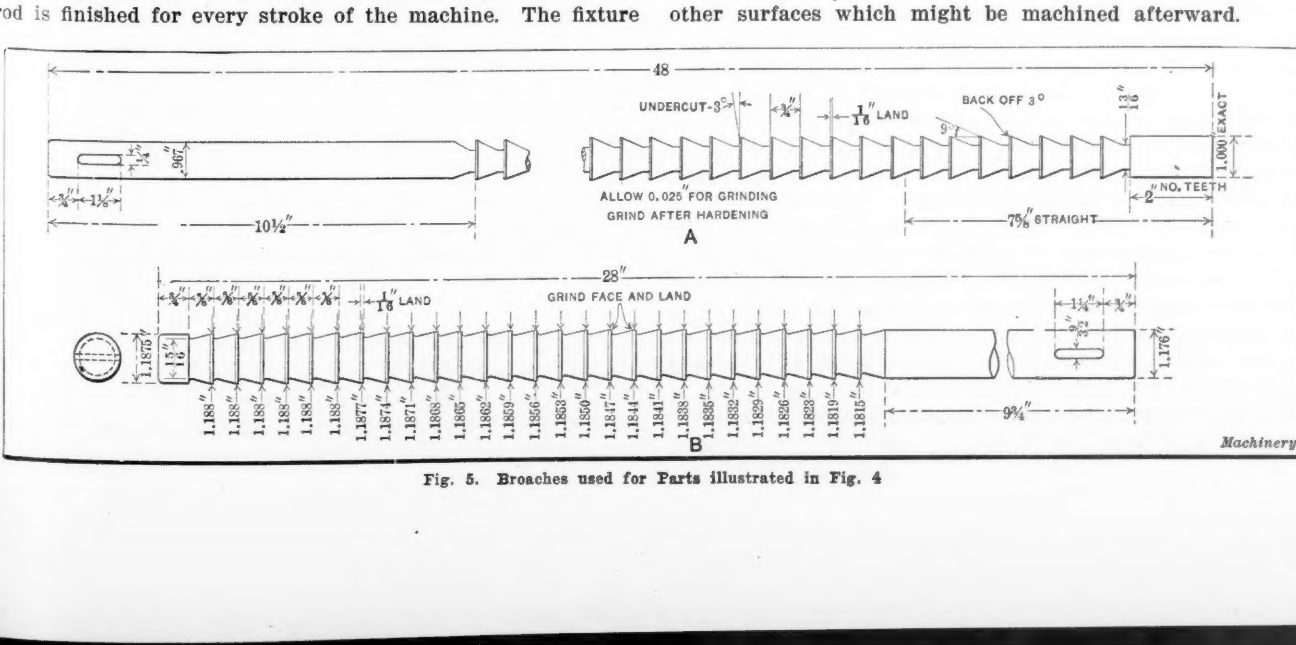


Fig. 5. Broaches used for Parts illustrated in Fig. 4

The part shown at A in Fig. 4 is another example of work on which broaching saved considerable time. This is a steering pivot for the front axle of an automobile. The hole is 5⅛ inches long and 1 inch in diameter. These pivots, at one time, were machined by first drilling the holes with two drills of different diameters which were followed by a rose reamer and a machine reamer, but even with these four operations, the hole was not finished straight and a hand reamer had to be used to secure satisfactory results.

Prior to the broaching operation, the pieces are roughly drilled 1/32 inch under size and then they are broached. The result is that the work does not need hand reaming and is produced cheaper and with a better finish than could be obtained by hand reaming. In this case, the broaching was done before any of the other machining operations. The material is chrome-nickel steel. The broach used for this work is shown at A in Fig. 5.

The example of work shown at B in Fig. 4 is another automobile part that is made of vanadium steel. The average production of these parts by broaching was sixty per hour, and one broach machined as many as 7685 pieces before it was worn too much for standard work. The dimensions of the broach used are given at B in Fig. 5.

* * *

CENTRIFUGAL FORCE ON AUTOMOBILE TIRES*

At first glance it would seem that every element going to make up the running gear of the modern automobile is balanced to such a nicety that any speed attainable under normal conditions would hardly be sufficient to necessitate serious consideration being given to any small, unbalanced weight. However, it has been demonstrated that even such a small weight as the tire valve is subjected to sufficient centrifugal force at high speeds, especially in racing cars, to warrant the placing of an equal weight on the opposite side of the wheel to overcome the energy developed by the rapid motion of the unbalanced valve.

The valves, including their caps, vary in weight for the different sized cars on which they are used from 3½ to 8½ ounces, and these weights seem insignificant when compared with the weight of a wheel and rim. But when the wheel is revolved at several hundred revolutions per minute a considerable centrifugal force is exerted, acting at any point in its travel radially away from the center of the wheel. It is intended to show here the tremendous forces that are developed at different speeds if the valves are not counterbalanced by weights on the opposite sides of the wheels.

The velocity V in feet per second of any particle of matter moving in a circular path with a constant radius, is represented by the product of the circumference and the number of revolutions per second. Now the centrifugal force of the same particle is the product of the weight and the square of the velocity, divided by the product of the radius and the force of gravity, which is 32.16 feet per second. Then substituting the value of V^2 in this second formula, we derive the equation, $C = 0.00034\ WRN^2$ pounds, where W is the weight of the body, R the radius of action, N the revolutions per minute and C the centrifugal force.

For the purpose of demonstrating by means of this formula, the necessity of balancing the wheels on high-speed cars, values will be assumed to correspond as nearly as possible to actual practice. Let the distance from the center of the wheel to the center of gravity of the valve be 15 inches; the weight of the valve and cap, 7½ ounces or 0.468 pound; and the diameter of the tire, 36 inches. Cars which will not attain a speed of 60 miles per hour are unusual, and it is an everyday occurrence for racers to maintain a speed of 75 or 80 miles per hour throughout a long contest or trial run. As a speed that is not at all unusual, let us assume 70 miles per hour for the first example. This is equivalent to 6160 feet per minute, and in traveling at this speed a wheel with a 36-inch tire will make 654 revolutions per minute. Substituting these values in the above equation we get:

$$C = 0.00034 \times 0.468 \times 1.25 \times 654 \times 654 = 85.1 \text{ pounds.}$$

In other words, a 7½-ounce valve on a 36-inch wheel traveling at 70 miles per hour will exert a lifting force of over 85 pounds when the valve reaches the top of the wheel. In case both valves of either pair of wheels are in the same relative position with regard to the axle, they will exert a combined lifting force of 170.2 pounds, and if they are opposite each other there will be a seesawing force acting on the car. Considering that there are four of these valves it is easily seen that they will exert forces in various and constantly changing directions as the wheels shift their relative positions in rounding turns in the road.

Since the centrifugal force varies as the square of the speed it requires only a slight increase in the speed to make a large increase in the force exerted. For instance, if the car travels at 75 miles per hour the force is increased to 97.5 pounds, and at 80 miles, which is frequently attained, the force on each valve will be 111.3 pounds, while at only 40 miles the force is 28 pounds. In a car going at the fastest rate of speed yet attained by man, 142 miles per hour, the force exerted by the valve is nearly 400 pounds.

Considering these almost neglected forces it is easily seen that some cognizance of them should be taken the same as is done in designing balanced wheels of stationary machinery. That racing drivers are coming to realize what these neglected forces might mean in case of an accident and loss of control of the car, is shown by the fact that a famous English driver has equipped each of his tires with two valves instead of one, the valves being placed on opposite sides of the tires in order to provide a perfect balance. It is said that the car will run at 80 miles per hour and hold the road remarkably well, where before it skidded badly at high speeds and was very difficult to control. Many American drivers also balance their valves by placing metal weights on the rim or felloes of the wheels opposite the valves.

* From an article by C. E. Palmer in the "Scientific American."

* * *

MOVING PICTURES OF AUTOMOBILE BUILDING

The Ford Motor Co. of Detroit, Mich., has just finished and equipped a complete department for taking motion pictures in its plant showing how Ford cars are made. A portion of the basement under the main office building has been fitted up as a studio and demonstrating room. The equipment is complete in every detail, and contains in addition to the regular photographic equipment a motion picture machine which is used for demonstrating to dealers and others interested. The films are made here and will be sent around to the principal cities both in this and foreign countries for exhibition in regular motion picture theaters. This is the first attempt that has been made by automobile manufacturers to exhibit their product in this manner before the general public, and it is reasonable to assume that it will create an unusual interest in the already popular Ford car.

The object of the company in establishing this department was to interest dealers and prospective purchasers. When a group of Ford dealers visit the Detroit office they are shown through the plant, taken out and photographed in a group outside the office building, and then are given a luncheon, after which the motion pictures of manufacturing Ford cars are shown. This entertainment lasts the greater part of the afternoon, and in the evening the party is treated to a banquet.

Duplicates of the films showing the processes of automobile manufacture are made from the original films and are sent to the principal cities and exhibited in regular motion picture theaters. After they have gone the rounds, they are turned over to district dealers and shown from time to time as occasion requires.

* * *

The use of natural gas as fuel in steam boilers is very wasteful, as it will take from four to five times the volume of gas per horsepower-hour to produce power through the medium of steam as it would if the same power were developed in a gas engine.

LETTERS ON PRACTICAL SUBJECTS

We pay only for articles published exclusively in MACHINERY.

ONE-PIECE ARMATURE DISK TOOLS

Having read the interesting article in the January number of MACHINERY, on "One-piece Armature Disk Tools," I can hardly understand how good results can be expected from these dies. After twenty years' experience in designing and constructing tools of this kind, I fail to see how it is possible to punch perfect blanks concentric and free from burrs with blanking or follow dies. I would prefer the compound sub-press construction, as this method keeps the upper and lower dies in perfect alignment, and saves a great deal of time in setting up the die. We make the lower or blanking die of one solid piece of "Ketos" steel (we also use "Intra" oil-hardening non-shrinking steel), machine to the required sizes and mount the die on the faceplate of the lathe to be bored (not drilled) to size. The device for mounting the die consists of a double plate with studs, and pawl to index for the required spacing. We also bore the holes for the oblong slots in the shedder and punch pad, with the same setting of the mounting device.

The prongs or individual punches for the upper die we make separately and mill them to the desired shape with forming cutters. This method enables us to replace a punch that breaks or distorts in hardening. Using a shedder that fits these punches nicely, not only sheds the punchings, but also supports and keeps the punches from springing. In shedding blanks in this manner one can leave the holes in the lower die straight without any clearance, whereas the method described in the article referred to requires some clearance to allow the punchings to drop through the die, which roughs up the holes to some extent. The holes in the die also become larger as the die is ground down; consequently the punchings will have heavier burrs. The pilot pins on follow-type dies that are depended upon to locate the stock cannot do so exactly, since they are made smaller than the piercing punches in order to prevent them from pulling the

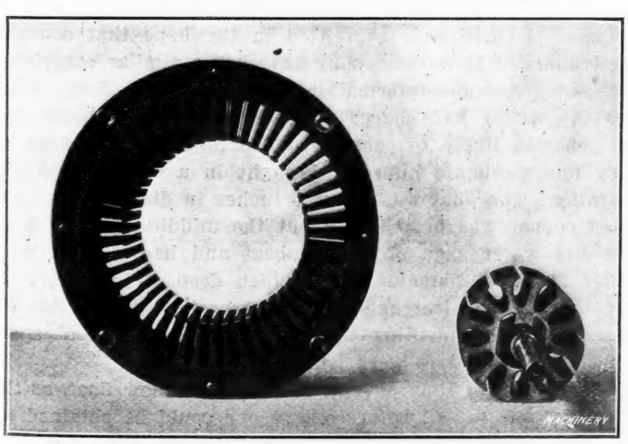


Fig. 1. A Series of Armature and Field Disks assembled—Note Perfect Alignment

blank out of the die; nor is the operator sure that the pilot pin enters the hole pierced centrally, so that it does not crowd the strip and thereby punch an imperfect blank. We are using black electrical sheet iron 0.015 to 0.019 inch thick, and make our punches 0.001 inch smaller than the dies for this thickness of stock, which gives good results. We make about fifty different sizes and styles of these, 1¼ inch to 14½ inches.

P. J. Mc.

[The chief advantage to be gained from making punches and dies from one piece of steel in preference to segments is the cheapness of construction. The method described in the foregoing criticism is the conventional one of making armature disk tools and contains nothing of unusual interest. The sectional sub-press type of armature punches and dies is not only difficult to make but is also extremely expensive. It was for this reason that the Robbins & Myers Co. decided to make its dies from one piece of steel and of the compound follow type. If this type of die is properly made and the press operator exercises the necessary care in setting up and operating, it is possible to produce blanks that are concentric and free from burrs. Fig. 1 shows this to good advantage. The series of field disks which have been clamped together look like one solid piece so exactly do the various segments conform to each other. By allowing only a small amount of clearance on the side of the die, it is possible to produce good clean disks until the dies are practically worn out.

As regards the toolmaking methods criticized in the above article, it is just as easy to machine a die in the milling machine as it is in the lathe, and, in fact, the milling machine is preferable if an accurate indexing head is obtainable. In the lathe it is necessary for the die to be trued up by means of an indicator which, of course, is only located by means of previously made prick-punch marks. Consequently, any inaccuracy in laying out will not be rectified in setting it up in

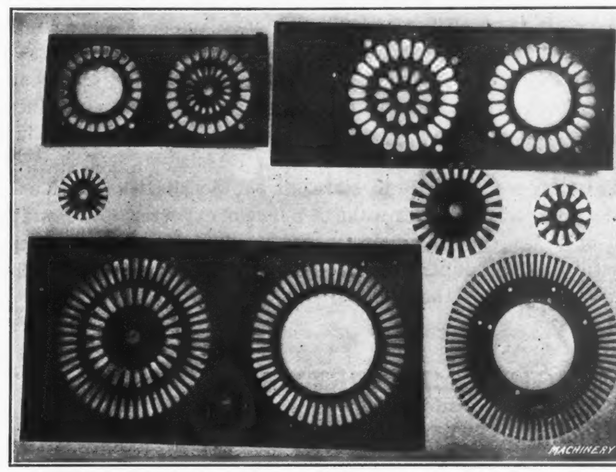


Fig. 2. Compound Punchings possible with the Solid Type of Armature Disk Tools

the machine to bore the holes. The Robbins & Myers Co. has been using the solid type of armature punches and dies for two and one-half years, and in that time has only had to repair two punches due to breakage of the prongs. In both cases repair pieces were set in with good results.

Another phase of the solid type of punch and die that is interesting is the production. This is considered to be greater than can be obtained with the combination sub-press type. With the solid type it is possible to get an armature or field disk, as shown in Fig. 2, at every stroke of the press and also to start the operation on another blank. This is impossible in the sub-press type because of the sectional type of construction used, and the prohibitive first cost. The solid type of armature disk tools is a good example of the advances made in the toolmaking art.—EDITOR.]

POSSIBILITIES OF RUSSIAN TRADE

The writer would like to add a few words to the item on page 379, engineering edition, of the January number of MACHINERY, concerning Russian export trade. It should be realized that, at the present time, Russia is recuperating from the effects of the Japanese war, and that great opportunities for American trade lie in the fact that this great country is keenly desirous of catching up with the progress of the rest of the world. To begin with, there are great expenditures for military purposes offering opportunities for trade. The government is, at the present time, building eleven large warships and twenty smaller ones. The army is very active in the development of aeroplanes and other military devices.

Russia is not so extremely backward a country as many would make us believe. There are thousands of investors and merchants familiar with developments in mechanical and other trades in other countries who are willing to take steps to develop the resources of their own country, but it is necessary that American manufacturers reach these people directly, as they do not yet seem to have the necessary push to take the initiative themselves. Advertising and direct methods for obtaining trade are necessary. Very few people seem to realize the rewards that may be waiting for the pioneers of American industries who enter Russia. The only serious competition that can be encountered is from Germany. Sweden can hardly compete with the United

States on the Russian market on account of its limited number of products*

As an example of the undeveloped state of Russia and the possibilities for further development might be mentioned the fact that there are only 120,000 miles of telegraph and 318,500 miles of telephone lines, these having been mostly constructed by Belgian and Swedish enterprises. The opportunities offered by this field alone are enormous. Russian water powers are also practically undeveloped, and there are great possibilities for turbines and electrical machinery. American agricultural machinery is used mostly on the larger estates at the present time, while the middle-class farmers hardly know of the existence of this class of implements. In addition, there are great possibilities for machinery for flour mills, beet sugar refineries, distilleries, breweries, and for the tobacco trade.

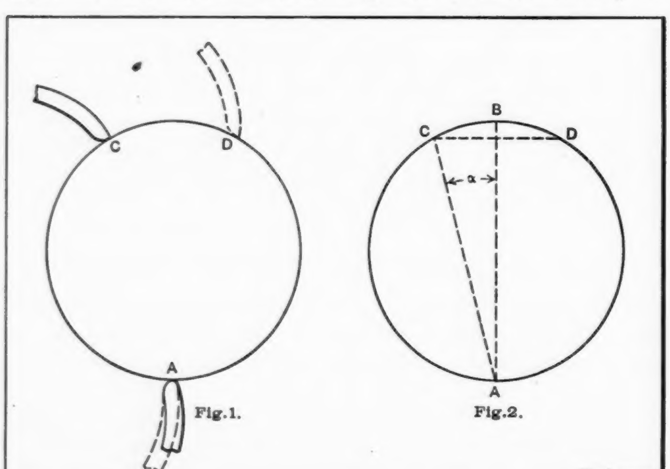
Several far-seeing concerns reached out early for the Russian market and profited thereby. As an example may be mentioned typewriters, small arms, and sewing machines. These have created a fine reputation for American products, and some firms, in fact, have had so large an amount of business that they have built factories in Russia.

It is, indeed, difficult to state all the possibilities that exist in Russia for the wide-awake American exporter. The writer can only say that if anyone has a product that has been successful in this country, he should go to Russia and try to find the same success there; also, now is the time!

L. J. W.

JUDGING MEASUREMENTS BY "DRAG" OF CALIPERS

It is evident that calipers set to 10 inches will not pass over a piece having a diameter of 10.010 inches without springing. That the distance *CD* in Fig. 1, which represents the "drag" of the calipers, depends upon the relation between the diameter of the work and the caliper setting, is equally self-evident. Probably most men who use lathes, especially when turning quite large work, sometimes rely on



Figs. 1 and 2. Diagrams illustrating how Allowances are judged by Drag of Calipers across Surface of Work

the drag of the calipers to show how much metal remains to be removed, and many veterans in the trade remember when allowances for press fits, etc., were made in this way, before the day of micrometers and vernier calipers.

I remember a man who was turning a part, the finished diameter of which was to be 20 inches. He roughed it down until the calipers dragged about 1/4 inch; then setting the tool in 0.005 inch, started the finishing chip in serene confidence that all was well. When the foreman, exasperated by the spoiling of a somewhat valuable piece, had concluded a full and free expression of his sentiments, the lathe hand quoted in his own defence what he believed to be a fact indisputable as the laws of gravitation. "Everyone knows, Mr. A., that the drag is always twenty times the oversize." As in this case the drag corresponding to 0.010 inch oversize is not 1/4 inch, but about 1 1/4 inch, his statement failed to meet with enthusiastic approval.

DRAG OF CALIPERS FOR DIFFERENT DIAMETERS

Caliper Setting, Inches	Drag for Oversize of			Caliper Setting, Inches	Drag for Oversize of		
	0.005	0.010	0.020		0.005	0.010	0.020
6	0.490	0.692	0.977	18	0.846	1.199	1.696
7	0.529	0.748	1.056	20	0.892	1.256	1.787
8	0.565	0.799	1.129	22	0 934	1.325	1.875
9	0.599	0.848	1.198	24	0.970	1.387	1.959
10	0.632	0.894	1.264	30	1.091	1.551	2.193
12	0.693	0.979	1.384	36	1.197	1.693	2.398
14	0.748	1.057	1.495	42	1.299	1.841	2.594
16	0.797	1.131	1.599	48	1.382	1.969	2.774

Machinery

It is easy to compute the distance *CD* for any given case. Referring to Fig. 2, $\frac{AC}{AB} = \cos a$, and $CD = 2\ AC \sin a$. In the instance cited, where the calipers were set to 20 inches, if the job were 20.010 inches,

$$\frac{20}{20.010} = \cos 1 \text{ degree } 48 \text{ minutes, and}$$

2×20 inches $\times$ sin 1 degree 48 minutes $= 1.256$ inch.

New London, N. H. GUY H. GARDNER

REMOVING A DENT FROM A COPPER FLOAT—A QUESTION

This contribution is submitted in the hope that some of the readers of MACHINERY may have had a similar experience and can give some information on the subject. Some years ago the writer had charge of a tool-room, and almost any old job was likely to come in. The foreman in charge (a very fine mechanic himself) brought in a copper float one morning. The float was about 5 inches in diameter and the sheet copper was brazed at about the middle. It was from the dial water gage of a steamboat and had a dent in it about 1 1/2 inch diameter and 3/4 inch deep. There were no openings in the piece and the trick was to take out this depression without drilling the hub or taking the float apart where the two halves were joined.

There was not time enough to send for a new float, as the boat was due to sail before a new one could be obtained. I was given no information as to how the work should be done and, in fact, considered that a joke was being played on me. The float was thrown on the bench and I had forgotten about it until the foreman finally called for it; then I realized that he was not joking. Well, we held a consultation but arrived at no solution and the foreman left me to figure it out. Here is the result.

The float was covered with a coating of lime about 1/32 inch thick, deposited from the feed water, and I picked up a small hammer that was used for laying out work and started to crack off the scale by way of amusement. This had not continued long before I imagined that the depression was smaller than when I started. Then I began to peen systematically around the depression, and lo and behold the dent was actually disappearing and in a short time was completely removed.

Mr. Foreman was very much surprised when the job was handed to him, and as he had said that "it was up to me" and I could worry about it, I made him guess a long time before I finally told him how the job was done. Now, I should like

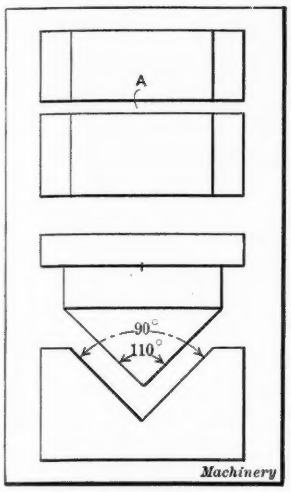
* Swedish competition is, perhaps, not to be so lightly disposed of. Sweden has made complete installations of telephone systems in a great number of the larger Russian cities in competition with the leading manufacturers both of Germany and the United States, and even nearer home it has offered United States formidable proof of its competitive qualities, in that a Swedish concern installed the complete telephone system in Mexico City in competition with manufacturers in the United States. Sweden, at the present time, and for a great many years back, has supplied Russia with most of the heavy wood-working machinery, such as saw mills, etc. used in that country, and many Swedish concerns are now establishing branch factories in Russia in order to overcome the difficulties caused by the Russian tariff. The only apparent reason why the Swedish competition might not, in the long run, be of importance would be because of the size of the country, it being too small in population to be able to supply the demands of a constantly increasing Russian market. The Swedes, however, are wide awake to the great opportunities in Russia, which many American exporters are not.

to have some of the readers of MACHINERY explain through what process the dent in this float straightened out.

Dubuque, Iowa. E. J. BUCHET

ANGLES OF ANGLE-BEAM SHEAR BLADES

J. D. Y.'s inquiry in November MACHINERY concerning angles of angle-beam shear blades calls to mind my experience in this line of work. Having described this in a contemporary some time ago, I will not go into much detail; however, I will say that by increasing the included angle of the upper or cutting blade to about 110 degrees and retaining the angle of 90 degrees at the bottom, very satisfactory results will be obtained. The reason for this is that the cutting is done gradually, beginning at the outer flanges of the angle, thus making it easier on punch, die and punch press. A further suggestion would be to do the cutting in the center *A* of two dies, as shown in the illustration, using a blade of a thickness depending on the weight and thickness of the angle to be sheared. This form of shear is excellent for this class of work and gives a very small amount of distortion in the angle.

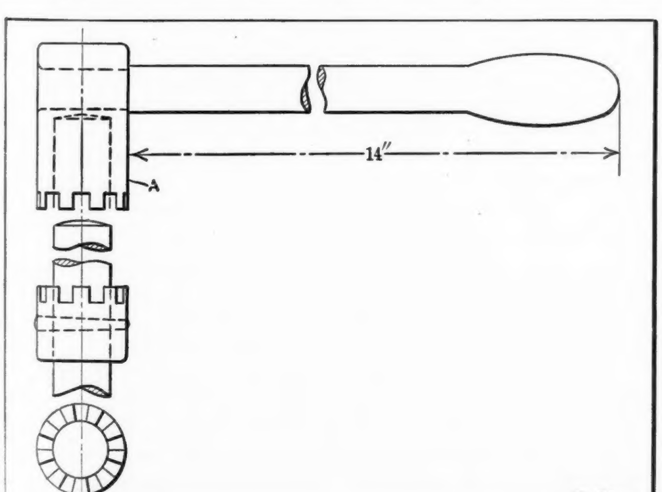


Angle for Angle Beam Shear Blades

New Haven, Conn. J. M. HENRY

HANDLE FOR CROSS-SLIDE OF TURRET LATHES

A very convenient handle for the cross-slide of a turret lathe or screw machine handling bar work is illustrated herewith. This handle is adjustable, and is therefore superior to the one sent with the machine, which is not adjustable. As the illustration shows, it is provided with clutch teeth so that by simply lifting it out of engagement the handle can be swung to any position. When two cross-slides are being used, the adjustable handle has a great advantage over the solid type. For example, if one slide is being used for cutting off finished parts while the other carries the forming tools, solid handles may interfere with each other, thus making it necessary to extend the forming tools so much as to cause chattering.



Handle for Cross-slides having Adjustment to prevent Interference between Handles on Front and Back Slides

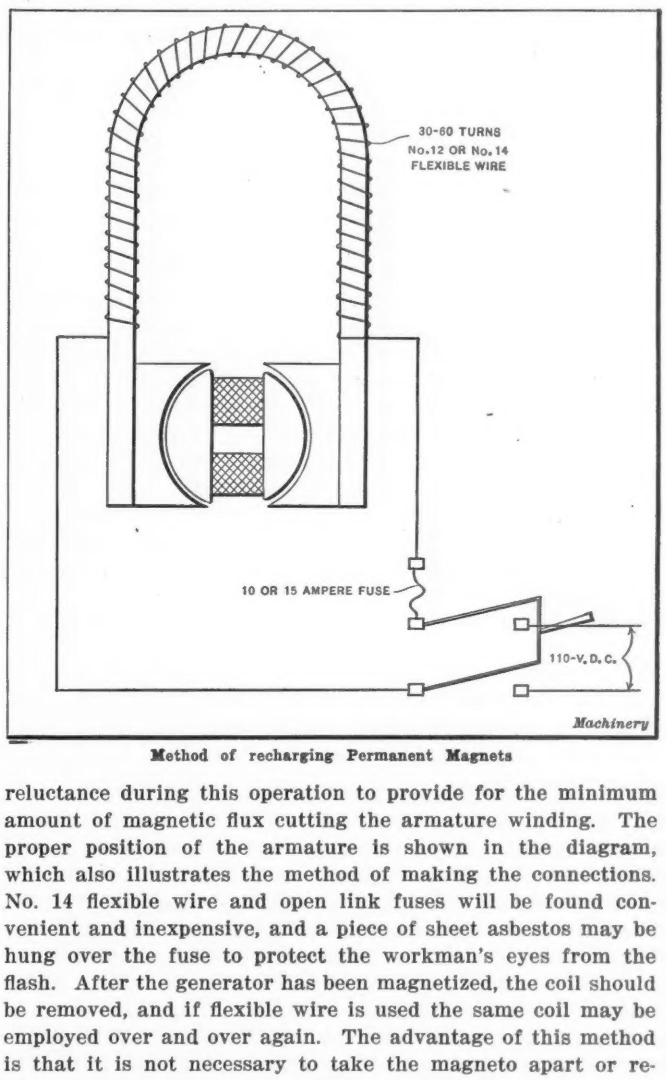
With the adjustable handles, such interference could be avoided. Another good feature is that the handle can be turned to the position that is most convenient for the operator and that enables him to exert the greatest pressure. The writer has designed and used this handle and found it a great labor saver. A handle of the solid type can be used in making the adjustable style, by simply cutting it off close to the hub ring and inserting the end in hub *A*. The particular handle illustrated has nine teeth (cut to a depth of 1/4 inch) which gives all the adjustment that is required.

Pearl River, N. Y. EDWARD A. FRIEL

RECHARGING PERMANENT MAGNETS

The December number of MACHINERY has just come to my desk, and if not too late I would like to offer an answer to A. B.'s inquiry in regard to a method of remagnetizing magneto generators—such as are used on automobile and motor cycle engines. These generators can be remagnetized by wrapping them with a temporary coil of from 30 to 60 turns of wire, and connecting this coil to a 110-volt direct-current circuit of considerable ampere capacity through a 10 or 15 ampere fuse and a knife switch. After the connections are made, the switch is closed, throwing the coil directly onto the line.

The fuse will be blown with some violence and the high current will force a heavy flux through the magnetic circuit and the armature. The armature, which is usually of the H-form, should be blocked in the position of least magnetic reluctance during this operation to provide for the minimum amount of magnetic flux cutting the armature winding. The proper position of the armature is shown in the diagram, which also illustrates the method of making the connections. No. 14 flexible wire and open link fuses will be found convenient and inexpensive, and a piece of sheet asbestos may be hung over the fuse to protect the workman's eyes from the flash. After the generator has been magnetized, the coil should be removed, and if flexible wire is used the same coil may be employed over and over again. The advantage of this method is that it is not necessary to take the magneto apart or remove it from the machine. This also eliminates the possibility of replacing the magnets with the polarity reversed.



Method of recharging Permanent Magnets

L. M. D.

SHRINK VS. PRESSED FITS

In answer to J. B. F.'s question in the December number of MACHINERY as to which is best, a shrink or a press fit, I would submit the results of the following experiments:

In making the rear axles of a well-known automobile, we were confronted with the question as to which was the stronger, a shrink or a press fit. This question was an-

swered by experiments as follows: The end members, brackets and housing had to be fastened securely to the steel tubing. First we tried shrinking the parts together. The tubes were turned to 2.254 inches diameter and the holes in the various members were reamed to 2.250 inches diameter and shrunk onto the tubes. Next we turned some tubes to 2.2525 inches, and the holes were reamed the same size as for shrinking, or 2.250 inches diameter. The members were then forced onto the tubes and, in both cases, were riveted with the same number of rivets. We made four axles with press fits and six with shrink fits. The axles were assembled and marked for identification and sent to the finish assembly department where they were fitted to the bodies.

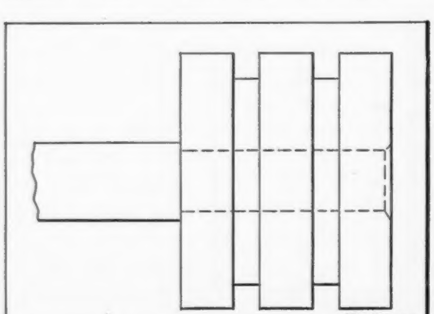
To make the story short, of the ten axles finished four were returned in less than a month and they proved to be the forced fits in each case. We never had an unfavorable report from the shrunk axles, and thereafter shrunk all parts onto the tubes and never had any complaints. The returned axles all had sheared rivets in the housing end and, in one case, the end member had sheared and was loose. This experiment was conducted with malleable iron castings. The outside diameter of the various parts was approximately 3 inches, leaving a wall about ⅜ inch thick. We afterward used Parson's white bronze, shrinking the members with the same allowance and less heat and obtained the same satisfactory results.

New Britain, Conn. W. C. Betz

Concerning J. B. F.'s problem of shrink and force fits, I would like to say that, theoretically, they would both be equal, provided the conditions were equal at the start. Theoretically, the holding power is limited by the elastic limit of the material which, in this case, would be the elastic limit of the outer member, as it is thinner than the shaft. In practice, a shrink fit will hold more firmly than a force fit because the gripping strain is equally distributed over the entire surface, while with a force fit the strain is unequal from one end to the other. This is due to the fact that the difference in diameter between the two parts must be taken care of by the compression of the shaft and the expansion of the outer member, and as the metal is not perfectly elastic, the end that goes on first will permanently set the shaft to a certain extent, so that the remainder of the fit will not grip tightly, owing to the diminished size of the shaft.

Detroit, Mich. Paul P. Vlasek

In the December number of Machinery J. B. F. asks a question in regard to the relative merits of shrink and pressed fits. In my opinion the collar which he illustrates in connection with his question should be shrunk onto the shaft. This method will be found most satisfactory where the collar or other part is relatively thin, so that there is danger of straining the metal in pressing it onto the shaft. Where a thicker piece, such as a crank disk, piston or similar part is to be secured to a shaft or rod, it is more practical to employ a pressed fit. It is a good plan to have a gage on the press so that it is possible to determine the exact pressure employed in forcing the piece into place. This insures having the fit tight enough to stand the work for which it is intended.



Small Piston to be shrunk or pressed on Rod

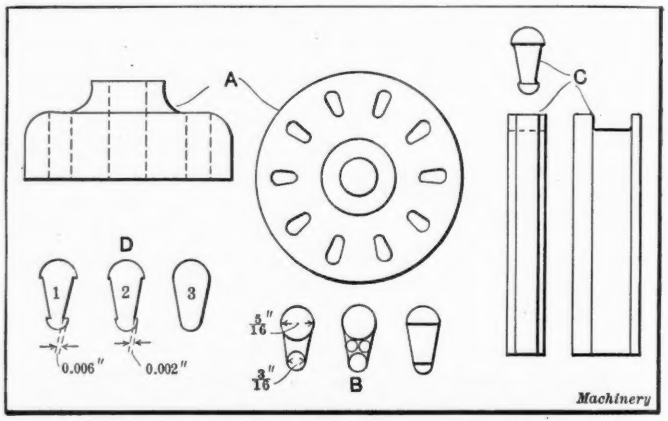
In this connection I would like to ask the most practical way of holding an 8-inch piston on its rod. Is a pressed fit or a shrink fit more likely to give satisfactory results? Will the fit become looser after the engine has run for some time and will the strain and vibration have any effect? I would also like to know how important it is to keep the rod or shaft cool while the piston or collar which is being shrunk onto it is cooling off.

W. J. M.

A BROACHING OPERATION

The method of broaching explained in the following paragraphs was worked out by the writer for machining ten holes in the core for a mold used for die-casting a small motor frame. These holes in the main core house sub-cores, as it was found impossible to produce perfect work and operate the mold in any other way. The main core is screwed to the top die, while the elongated sub-cores are fastened to a core plate. The main core is freed from the casting after the mold is separated, by the action of ejector pins, after which the sub-cores are withdrawn.

The broaching of these holes was performed by three broaches, the second and third of which varied in size by only 0.002 inch. The purpose of this was to have the third broach take a very fine cut so that the desired finish would be obtained without any other work than a small amount of filing for clearance at the back of the hole, and polishing with emery cloth. It was also important that the edges of the sub-core holes on the face of the main core should be as nearly perfect as possible; otherwise a burr would form on the casting at this point.



The Work to be broached and the Type of Broach that was used

It will be seen that these core pin holes pass through the core *A* as shown in the accompanying illustration. The core is made of machine steel and is 1½ inch thick. The core pin holes are ⅝ inch long, the maximum width being 5/16 inch and the minimum width 3/16 inch. The ends of the holes are semicircles of 5/32 and 3/32 inch radius, respectively. In laying out the work, two concentric circles were scribed on the core *A* at the required distances from the center. The next step was to draw ten equally spaced radii to locate the centers of the holes to be broached. The centers of the 5/16 and 3/16 inch semicircles were next marked with a center punch at the intersections of the two concentric circles with these radii. It may be well to mention at this point that all other machining operations had been performed on the core before starting to machine the core pin holes.

The next step was to drill ten holes of 5/16-inch diameter and ten holes of 3/16-inch diameter on the centers which were located for this purpose. These holes were then reamed. This left a wall of metal of ⅛-inch thickness between the two holes and this wall was partially removed by drilling two small holes as shown at *B*. The remainder of the wall was then broken away by pushing a cold chisel through the hole, after which the holes were ready for the broaching operations.

Three broaches were made of the form shown at *C*, the broaches being about ½ inch longer than the thickness of the core *A*. The broaches were made a few thousandths under size to provide a running fit against the remaining portion of the walls of the 5/16 and 3/16 inch holes, which served as a guide for the broaches. The cutting end of each broach was filed away to a depth of 3/16 inch at the center, leaving segments of 5/16 and 3/16 inch diameter on the broach which

served as a guide in starting it into the hole and leading it through the work.

The broaches were not stepped in the usual way, but were designed to simply cut on the end, the form of the three broaches used being shown at *D*. Broach No. 1 was cut away to a depth of 0.006 inch on each side. Broach No. 2 was cut away in a similar manner to a depth of 0.002 inch on each side. Broach No. 3 was not cut away on the sides. It will be evident that broach No. 1 leaves 0.006 inch of metal on the sides of the hole, and that broach No. 2 cuts away 0.004 inch of this metal, leaving 0.002 inch for the finishing cut taken by broach No. 3. An arbor press was used to force the broaches through the work, plenty of oil being supplied for lubricating purposes. Smooth clean holes were produced that were uniform in size and required very little finishing. The cost of the machining operation was much less than it would have been if each hole had been filed to a plug gage.

Union Hill, N. J. G. I. Johnson

THE SQUARE KEY VS. RECTANGULAR AND TAPERED KEYS

In an article on keys which appeared in the November number of Machinery, Martin H. Ball hoped to start a discussion of the subject. It is an interesting subject and it is a pleasure to me to see someone else make a start on it. Seeing the key situation in a different light does not necessarily make me right or the other fellow wrong; so let us have all the light we can. The square key, without taper, made of cold-rolled stock, fills the bill completely from my point of view. It can be bought finished to size and in convenient lengths so that it can be kept in stock. Then when a key is wanted in a hurry, as is often the case, sawing to length is all that is necessary. If the square key should be recognized as the universal standard, all squabbles as to the proportion of width and thickness would be done for. Any number of combinations can be made up for a standard of rectangular keys, but square is square all over.

Mr. Ball states that customers complain of square keys, saying that they unnecessarily weaken the shaft and hub; and broken parts returned seemed to justify their conclusion. The fact that parts were broken does not prove that the key was at fault. The shaft may have been too weak to do its work properly, and the same may be true of the hub. That is too often the case. If shafts, keys and hubs were in all cases properly designed for their loads instead of by rule-of-thumb methods, we would not hear of so many cases of hubs bursting or keys shearing. In many of my designs I have used standard safety flange couplings, but when making computations for hub and key I have often had to use two standard keys in order to keep the key stress within safe limits. And often when using a pinion that is of small diameter in proportion to the shaft I find it advisable to use two small keys in order to keep the hub stress low. In Kent's "Pocket-book" we find that common practice is to use a square key whose width and depth are each equal to one-fourth the diameter of the shaft, or as nearly as may be in even sixteenths of an inch. Ordinarily that does very well for the section of the key, but it is still necessary to make a computation to find out whether one or two keys are needed and how long they should be.

Everyone that is interested in this subject should send to the University of Illinois, Engineering Experiment Station, Urbana, Ill., for Bulletin No. 42. This bulletin contains the results of tests on numerous shafts with keyways of common sizes. On page 10 we find this significant statement, "It seems that a shaft with a single keyway of common dimensions has about the same ultimate strength as a shaft without a keyway." The statement in regard to the weakening effect of two keyways of standard dimensions must not be confused with the statement made in a preceding paragraph in regard to the advisability of sometimes using two keyways of smaller section in place of one standard keyway. To the best of my knowledge no tests have been conducted to determine the effect on the strength of a shaft of two keyways whose shearing value is equal to one keyway of standard dimensions. I have frequently used two keys in a pinion in order to keep the hub stresses down, and, of course, this means that the keys must be smaller than the standard sizes. Under such conditions I do not think the shaft is any weaker than it would be if one keyway of the regular size were cut in it. So far I have had no complaints in regard to this practice.

As an example of the use of two small keyways, consider the case of a pinion whose hub diameter is 6 inches and the shaft diameter 4 inches. The regular square key would be 1 inch thick and the remaining section of the hub left to resist tension would be only ½ inch thick. If two keys were used, each of which were ½ inch thick, the resistance of the keys against shear would be the same, and the hub would have a net thickness of ¾ inch, which represents a gain of 50 per cent. Although it is not definitely known, it appears very doubtful if the shaft has been weakened any more through cutting two ½-inch keyways than it would have been by cutting one keyway for a 1-inch key.

Experience has shown that a taper key will often produce considerable stress in the hub, tending to burst it. And also a taper key has been known to spring even quite heavy members so much as to make the machine a failure. Similar members when keyed with a square key have given a successful machine. To me, a taper key appears to be nothing more than a convenient means of tightening up a poor fit at the expense of the rest of the job. On investigation, it will be found that many of our largest makers of machinery are using the square key as their standard. The General Electric Co. might be mentioned as one of them.

Albany, N. Y. Charles P. Wiweke

SEMI-AUTOMATIC TURRET FIXTURE

It was necessary to cut a fine thread on work of the form shown in Fig. 1, and as the preceding operations were performed on these pieces in a turret lathe, it was advisable to do the threading on the turret lathe also. None of the stock feed gears would produce the necessary result, and rather than make a special gear or send to the manufacturer for one, the fixture illustrated in Fig. 2 was designed. This fixture has proved not only practical, but even more economical than if the operation had been done with the feed gears and cam. Of course the device is only semi-automatic, but the threading is the last operation performed and the machine requires the attention of the operator at this point.

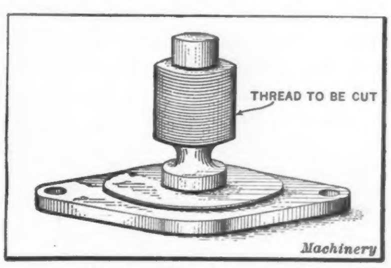


Fig. 1. Work to be threaded

It will be noted from Fig. 2 that the fixture is a sort of

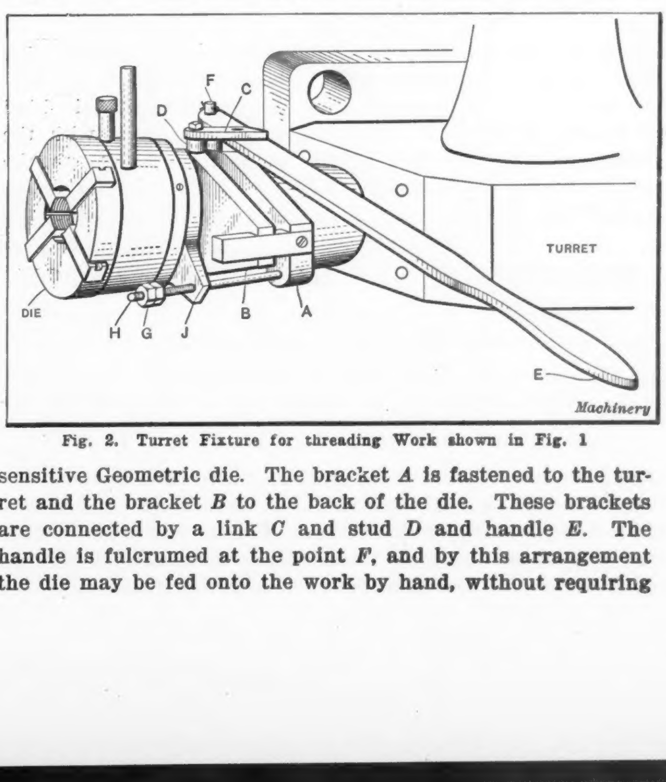


Fig. 2. Turret Fixture for threading Work shown in Fig. 1

sensitive Geometric die. The bracket *A* is fastened to the turret and the bracket *B* to the back of the die. These brackets are connected by a link *C* and stud *D* and handle *E*. The handle is fulcrumed at the point *F*, and by this arrangement the die may be fed onto the work by hand, without requiring

undue effort on the part of the operator. The turret is stationary at the end of its stroke, which does not bring the die into contact with the work, but close to it. The nuts *G* on the stud *H* may be adjusted to engage trip *J* at the desired point; this trip opens the die and allows it to be withdrawn quickly by moving back the hand lever *E*. An automatic device for resetting the die could easily be attached to the turret to make the tool ready for operation on the next piece of work.

RICHARD RUSSELL

DRILL PRESS FIXTURE

The following describes a drill press fixture which is used for holding work of irregular shape. It will be seen that the fixture consists of six bars which have shouldered studs mounted in them. The work is set on the shoulders of these

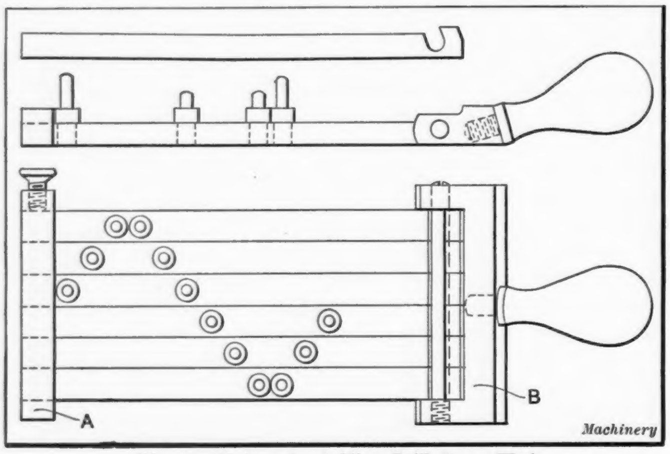


Fig. 1. Fixture for holding Drill Press Work

studs to provide clearance underneath so that the drill will not cut into the fixture. The method will be readily understood by referring to Fig. 2.

In preparing to set up a piece of work in this tool, the strap *A* and handle-piece *B* are removed. This leaves the bars free so that they may be arranged to bring the studs into the desired position, three or four studs bearing against the work as shown in Fig. 2. The handle-piece and strap

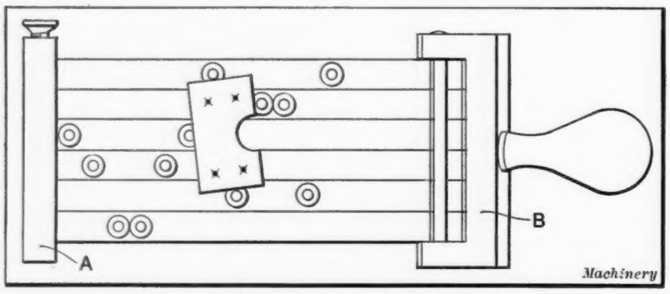


Fig. 2. Work in place in Drilling Fixture

are then replaced and the work is ready to be drilled. Before mounting in the fixture, the holes are usually centered as indicated by crosses in Fig. 2. A fixture of this kind is in the nature of a safeguard, as it does away with the danger of the operator's hands being injured through the work getting caught and being twisted around by the drill.

Kilfinane, Ireland. C. W. WHITESIDE

SLOTTING "NI-CHROME" STEEL RIBBON

In response to W. C. H.'s inquiry on slotting "ni-chrome" steel, in November MACHINERY, I do not think it is possible to produce a die that would satisfactorily slot "ni-chrome" steel ribbon. As the writer suggests, it is the toughest material to work that could be imagined. I have not had experience in this line exactly, but venture the following suggestion. In the making of the ordinary sub-treasury locks, the small flat disks that form the main part of the locking device, are slotted by means of disks of metal run at about 1500 revolutions per minute. The disks are not more than 0.100 inch thick. It was attempted at first to saw these slots, but a burr was left which made the work unsatisfactory. With just a plain metal disk, these slots are now formed, leaving practically no burr. This method was described in MACHINERY some time ago. My suggestion would be to build a machine with, perhaps, a dozen of these disks in line and a clamping arrangement to hold the ribbon down while it is being slotted, but, as I said before, this is merely a suggestion, although experiments may prove it to be of value.

New Haven, Conn. J. M. HENRY

ADJUSTABLE CLAMPS FOR T-SQUARES

In cross-sectioning drawings it is frequently necessary to keep the T-square in one position for a considerable length of time while using the triangle to cross-hatch some detail of the drawing. If the T-square slips, it means that a lot of time will be lost in erasing and then redrawing the lines over which the pen passed. To obviate this difficulty, the writer has used two forms of clamps which are illustrated herewith. The use of these clamps holds the T-square in place and leaves one hand free to use the drawing pen while the other can be used to move the triangle. In this way, quicker and better work can be obtained.

In order to use the clamp shown in Fig. 1, a slot 1 inch wide by 3/8 inch deep is cut along the left-hand edge of the drawing board. A strip of metal of the same size is then fastened to the edge of the board by means of countersunk screws, the metal being allowed to project 1/4 inch beyond the edge of the board. In this way, a slot 1/4 inch wide is left between the metal strip and the edge of the drawing board.

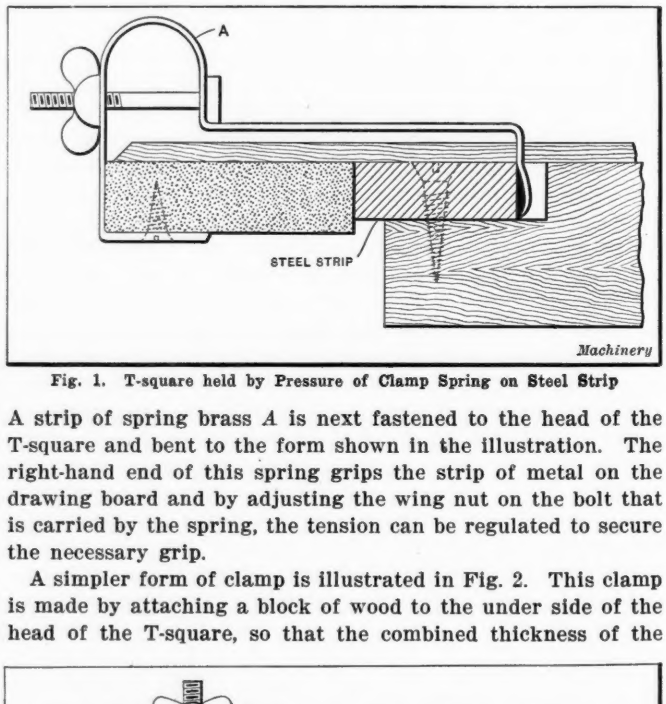


Fig. 1. T-square held by Pressure of Clamp Spring on Steel Strip

A strip of spring brass *A* is next fastened to the head of the T-square and bent to the form shown in the illustration. The right-hand end of this spring grips the strip of metal on the drawing board and by adjusting the wing nut on the bolt that is carried by the spring, the tension can be regulated to secure the necessary grip.

A simpler form of clamp is illustrated in Fig. 2. This clamp is made by attaching a block of wood to the under side of the head of the T-square, so that the combined thickness of the

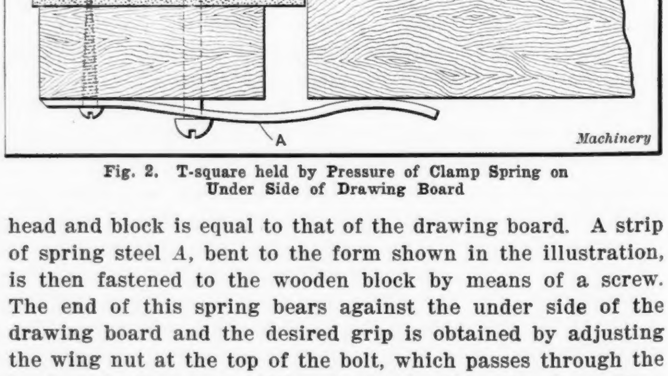


Fig. 2. T-square held by Pressure of Clamp Spring on Under Side of Drawing Board

head and block is equal to that of the drawing board. A strip of spring steel *A*, bent to the form shown in the illustration, is then fastened to the wooden block by means of a screw. The end of this spring bears against the under side of the drawing board and the desired grip is obtained by adjusting the wing nut at the top of the bolt, which passes through the head of the T-square. Although simpler than the clamp illustrated in Fig. 1, this arrangement does not work as well because the action of the spring sliding over the wood is not

as smooth as when the spring is in contact with a metal surface. Another disadvantage of this clamp is that it is impossible to hold the T-square as firmly with it.

Allegheny, Pa. MURRAY FAHNESTOCK

COMPENSATING SCREW THREADS FOR SHRINKAGE

In the December number of MACHINERY, W. B. T. presented an illustration of a chuck jaw which had shrunk 1/64 inch in pitch in hardening; the $1\frac{1}{2}$-inch diameter screw to fit this jaw, which has five left-hand square threads per inch, was also illustrated. Since tool steel shrinks only 0.015 to 0.030 inch per foot of length if it is correctly handled, it is assumed that W. B. T. meant that there was a shrinkage of 1/64 inch in the entire length of the jaw—say 4 inches. The method of compensation presented in the following is applicable, however, to any amount of shrinkage and may also be used in cases where it is desired to cut screws with an odd or fractional lead.

In the case under consideration, assume that in 4 inches the lead has shrunk 1/64 or 0.0156 inch, making the length of twenty threads 3.9844 instead of 4.000 inches. Under these conditions the lead of the screw will be $\frac{3.9844}{20} = 0.19922$ inch. Consider the case of a lathe which has a lead-screw with six threads per inch. It is then necessary to find gears with a ratio of $\frac{0.19922}{\frac{1}{6}}$ to drive this screw. Multiplying this fraction by $\frac{6}{6}$ gives $\frac{1.19532}{1}$. To further simplify this fraction, divide both numerator and denominator by 4 and then multiply numerator and denominator by 10,000 which gives the following result:

$$\frac{1.19532 \div 4}{1 \div 4} = \frac{0.29883}{0.25000}$$

$$\frac{0.29883 \times 10{,}000}{0.25000 \times 10{,}000} = \frac{29{,}883}{25{,}000} = \frac{7471}{6250}$$

This fraction cannot be easily reduced further and in order to bring it to a form where it can be conveniently handled, add to the numerator and denominator two numbers whose values are such that the resulting value of the fraction will remain unchanged. For example, if to both terms of the fraction $\frac{150}{300}$ numbers are added in the ratio of 1 to 2, the value of the fraction remains the same, thus:

$$\frac{150 + 7}{300 + 14} = \frac{157 + 9}{314 + 18} = \frac{166}{332} = \frac{1}{2}$$

This process can be continued indefinitely, provided the numbers added to the numerator and denominator are in the same ratio as that of the original fraction. The fraction $\frac{7471}{6250}$ is very nearly $\frac{6}{5}$ and so $\frac{4 \times 6}{4 \times 5}$ may be added to it without changing the ratio. Performing this step gives $\frac{7495}{6270} =$

$$\frac{1499}{1254} = \frac{1500}{1255} = \frac{300}{251} = \frac{306}{256} = \frac{6 \times 51}{8 \times 32} = \frac{30 \times 51}{40 \times 32} \text{ (approximately).}$$

The result of the preceding calculation shows that we may use gears having 30, 51, 40 and 32 teeth, respectively. The lead of the screw to be cut is equal to the lead of the lead-screw on the lathe multiplied by the ratio of the product of the number of teeth in the driving gears to the product of the number of teeth in the driven gears. In the present case, the gears with 30 and 51 teeth are the drivers and those with 40 and 32 teeth are the driven gears. This gives the screw to be cut a lead of

$$\frac{1}{6} \times \frac{30 \times 51}{40 \times 32} = \frac{51}{256} = 0.1992187 \text{ inch.}$$

This result differs from the desired lead by only 0.000002 inch, which is far too small to be easily measured even if the screw to be cut was one foot in length. Gears with 30 and 40 teeth will always be available among the collection of change gears for any screw cutting lathe and a 32-tooth gear will generally be included in the collection. This leaves only one gear having 51 teeth to be cut for this particular job.

As another example, assume a case in which the hardened jaw is 3 inches long and 15 threads are exactly 2.9844 inch in length after hardening, and that it is required to drive the lead-screw (five threads per inch) with gears having a ratio of $\frac{0.19896}{0.20000}$. Simplifying, we have:

$$\frac{0.19896}{0.20000} \times \frac{50{,}000}{50{,}000} = \frac{9948}{10{,}000} = \frac{2487}{2500}.$$

The value of this fraction is so nearly unity that the same number can be subtracted from each term without seriously affecting the result. The required driving and driven gears are thus found to be:

$$\frac{2487 - 30}{2500 - 30} = \frac{2457}{2470} = \frac{27 \times 91}{38 \times 65} = 0.994737.$$

This results in an error of less than 0.001 inch per foot.

As a further example, consider a case in which it is required to cut a screw with a lead of 0.19922 inch on a lathe whose lead-screw has a lead of 0.250 inch. The ratio then becomes:

$$\frac{0.19922}{0.25000} = \frac{9961 + 4}{12500 + 5} = \frac{1993 + 4}{2501 + 5} = \frac{1997}{2506}$$

$$\frac{27 \times 74}{23 \times 109} = 0.796968 \text{ (approximately).}$$

This gives an error of only 0.00011 inch per inch. These examples are sufficient to show that with a few odd gears and a lathe—preferably with a fine-pitch lead-screw—it is possible to cut any fractional thread with very small error. The error will frequently be so slight that it is much smaller than can be determined by measurement. The reduction of the fractions to the required form will be greatly facilitated if a list of prime numbers is available for this purpose. After a little practice a set of compound gears for any required lead can be picked out very quickly. The writer uses this method to chase tap threads, cutting the lead just enough longer than the required dimension so that the pitch of the thread will be exactly right after the tap has been hardened.

Boston, Mass. L. J. RODGERS

RECHARGING PERMANENT MAGNETS

In the December number of MACHINERY, A. B. asks how to recharge a permanent magnet. The following will be found a satisfactory method:

Attach the permanent magnet that is to be recharged to a direct-current electromagnet. Allow the two magnets to stay in this way for about 45 minutes and strike the permanent magnet light blows with a hammer every few minutes. The small particles of the steel which the chemist calls "molecules" must all lie in the same direction in order for the magnet to retain its magnetism. The light blows struck with the hammer while the current is flowing sets up a vibration in the steel which enables the molecules to adjust their position so that the magnetism is retained after the magnet to be charged is disconnected from the electromagnet.

Grand Rapids, Mich. GEORGE H. HAMILTON

BRAZING CAST IRON

Mechanics who are called upon to braze cast-iron parts together will find the following a very satisfactory method. Place the pieces to be brazed in such a position that the fractured surfaces are uppermost and then heat them slowly to a temperature of about 2000 degrees F. This is the highest temperature that can be safely used on cast iron without running the risk of melting it. After this temperature has been attained, the work is allowed to cool slowly. The parts are

next clamped securely together and brought to a bright red heat. A flux, composed of the following ingredients is then applied:

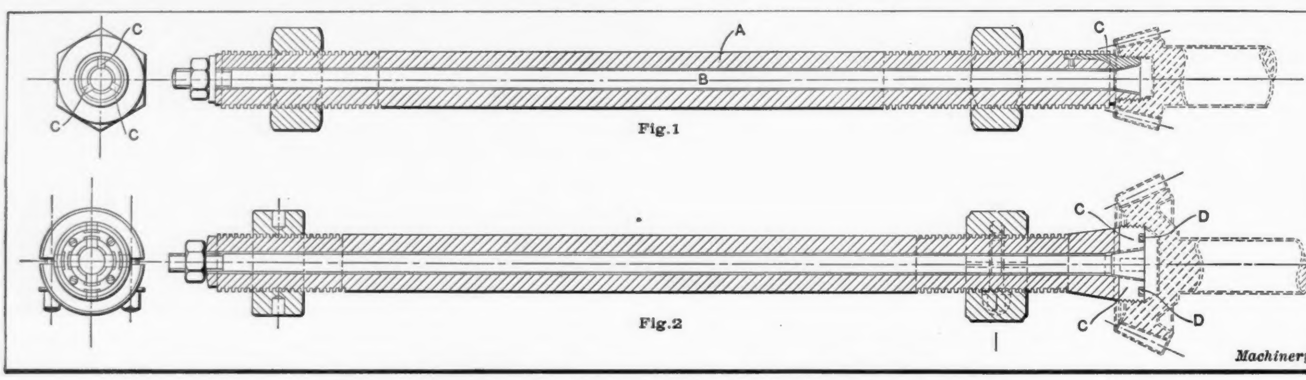
Iron carbonate................................1 part
Powdered boric acid........................2 parts

It is important for the parts to be brought to a bright red before adding the flux, which should be applied with a brass rod. After brazing, the work is allowed to cool in the air. In heating the work an ordinary gas torch is found very satisfactory.

East Orange, N. J. GEORGE GARRISON

SPECIAL ARBOR FOR BEVEL GEAR SHAPER

It is rather difficult to hold firmly in the Bilgram gear shaper, any pinion which has not a hole through it but is



Figs. 1 and 2. Expanding Arbor for holding Bevel Gears in Gear Generator

made solid with the shank as in the case of pinions for the rear drive of motor cars. For this reason, we have designed and have been using with success, the arbor illustrated in Figs. 1 and 2. Bevel pinions are bored and threaded, as shown to the right, so that they can be screwed onto arbor *A* which is supplied with the machine. This arbor has a hole through it to receive shaft *B*, one end of which is conical and the other threaded for a hexagonal nut. As will be seen, the end of arbor *A* which holds the pinion, has three or four notches in which are located strips or dies *C* that were threaded with the arbor. The inside edges of these strips bear against the taper end of the shaft *B*, and by drawing the shaft outward by means of the nut, the strips *C* expand, causing the pinion to be held firmly. Fig. 1 shows the arbor used for very small pinions. In this case, the expansion strips are held in place by set-screws which retain them in the arbor but allow some play. Fig. 2 shows an arbor suitable for pinions of larger size. There are four expanding strips held in place by ring *D* which is fixed to the arbor by set-screws.

Turin, Italy. C. BOELLA

FIXTURE FOR MILLING CONNECTING-RODS

The fixture for milling connecting-rods illustrated in Figs. 1, 2 and 3, was designed to enable work to be done in one operation on one machine that formerly required two operations on two machines. Only one operator is required for handling this work in either case, the saving effected by this fixture arising from the fact that the second machine can be used for some other work. The present method also produces more accurate work and the rate of production is increased.

The connecting-rods machined in this fixture are used on automobile engines. As they are heat-treated before machining, the metal is extremely tough but with high-speed steel inserted-tooth milling cutters, we were able to turn out from 275 to 300 connecting-rods in a ten-hour day. No time was lost in indexing the fixture or in removing the finished connecting-rods and replacing them with rough forgings, as the operator had ample time to perform this part of the work on one fixture while the cutters were at work on the rods held in the other fixture.

Figs. 1 to 3. Plan and Sectional Views of Fixture for milling Connecting-rods

Referring to the sectional view shown in Fig. 2, it will be seen that the fixture consists of a base casting *A*. A casting *B* is pivoted to the base casting and the locating V-blocks *C* and *D* are bolted to it. These V-blocks are mounted on pivots and allowed to swivel through a short distance, their movement being governed by stop-pins in the V-blocks which fit into holes in the casting *B*. This movement of the V-blocks is provided to compensate for any variation in the connecting-rod forgings, and keeps the work centrally located. The cover casting *E* is bolted to the top of the swivel casting *B*, and the clamping screws, levers *F*, posts *G* and clamping V-blocks *H* and *I* are secured to this cover casting. The clamping V-blocks are provided with the same swivel movement as the locating V-blocks, which insures the required alignment. Hardened steel pins *J* are inserted in the cover to take the thrust of the clamping screws *K*.

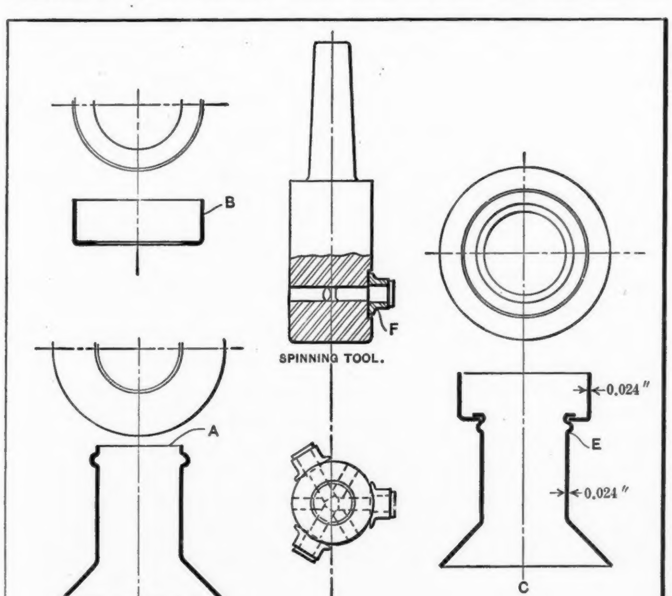
It will be seen that two connecting-rods are mounted in the fixture with their opposite ends adjacent, and the position of the straddle cutters is clearly shown in Fig. 1. The fixture is cut away sufficiently to allow the work to be fed to these cutters to machine the faces of the bearings. After one end of the connecting-rods has been machined, the lever *M* is pushed down to draw the index pin *N* free, and the casting *B* is then swiveled through 180 degrees and again located by the pin *N*. The fixture is then ready for the opposite ends of the connecting-rods to be machined. A gage, not shown in the illustration, is fastened to the base casting *A* to provide for setting the milling cutters.

Detroit, Mich. LOUIS L. ROBERTS

SPINNING TOOL FOR ASSEMBLING SHELLS

The brass shell *A* (see illustration) was to be attached to shell *B*. The regular way in which this class of work had been done was to make a split die, one-half being operated by a cam to open and close it, which allowed the work to be placed in the die. An undesirable feature of this method was that the shell was clamped, to support it, under the bead *E*, which caused the bead to be flattened.

The spinning tool *F* was devised for this work and has given good results. Besides retaining the shape of the bead, it makes a tighter joint and the production is increased over 75 per cent. The two shells, after being closed together by spinning, are shown at *C*. The spinning tool, which is used in a drill press, is run at about 500 revolutions per minute, and the vertical movement of the tool is obtained by a foot-treadle. The body of the tool is made of tool steel, hardened in oil and tempered at 495 degrees F. (brown). The rolls and pins are of tool steel, hardened in oil and tempered at 435 degrees F. (pale yellow). The rolls are an easy running fit



Spinning Tool and Die used for assembling Shells as shown at C

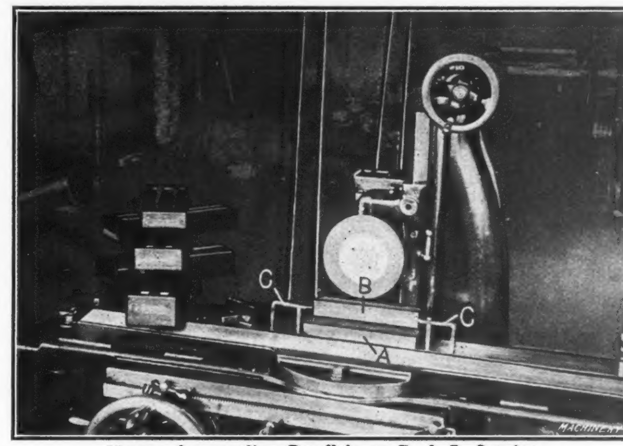
on the pins, and the pins a driving fit in the body. The die is made from cast iron and is bolted to the table of the press. The plug *D* is made of low-carbon steel and supports the inside of the shell. The shells are assembled, placed in the die and held from turning by the fingers, a slight pressure between thumb and forefinger being sufficient to hold the work while the tool does the spinning. This style of tool and die is now being used instead of the split die, except in cases where it is necessary to have the bead flattened so the inside walls will come together.

W. D.

SIMPLE GRINDING FIXTURE

The accompanying illustration shows a simple fixture that was used on a No. 13 B. & S. grinder to finish accurately five sets of hardened parallels. They are shown at the left-hand end of the grinder table in hard wood boxes, which keep them in pairs and protect them from injury when not in use.

In making the grinding fixture the base *A*, which was a gray iron casting, was secured to the table by means of two 5/16-inch fillister-head screws. The base was then ground



Fixture for grinding Parallels on B. & S. Grinder

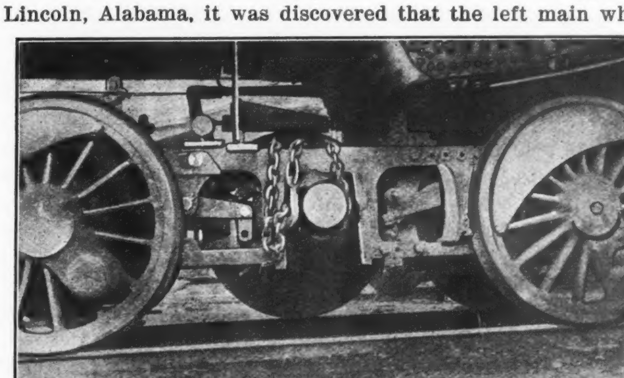
off to an accurate surface on its top face. The part *B* was next screwed to the base of the fixture and ground to an accurate finish so that its vertical face was perpendicular to the top of the base *A*. The clamps *C* were used to hold the work down and two parallel clamps were also used—one at each end—to hold the work securely against the face of the block *B*. These parallels were finished in pairs, and after being completed did not show an error of over 0.0005 inch in any direction. The sizes ranged from 3/8 by 3/4 inch to 3/4 by 1 3/4 inch, all sizes being 7 inches long.

Pottstown, Pa. CLAYTON DANE

A PECULIAR ACCIDENT

The accompanying photograph shows an accident that happened on the Birmingham division of the Southern Railway about a year ago. There are several cases on record of a pair of trucks having slipped from under a box car unnoticed, and one famous case where a whole car was lost out of a freight train without being detected until the end of the run.

The engine shown in the illustration had been stripped of all side-rods and was being hauled along with a train of cars from Atlanta to Birmingham. When the train stopped at Lincoln, Alabama, it was discovered that the left main wheel



Locomotive which lost Driving Wheel, Box, Shoe, Wedge and Pedestal Brace while being hauled in Freight Train

as well as the driving-box, shoe, wedge and bottom rest were all missing. There had been no previous indication that this wheel was loose. As is quite generally known, these wheels are pressed onto the axle with a pressure of from 100 to 150 tons, and, although they sometimes loosen after long usage, we never heard before of one working clear off the axle. The missing wheel was found twelve miles back.

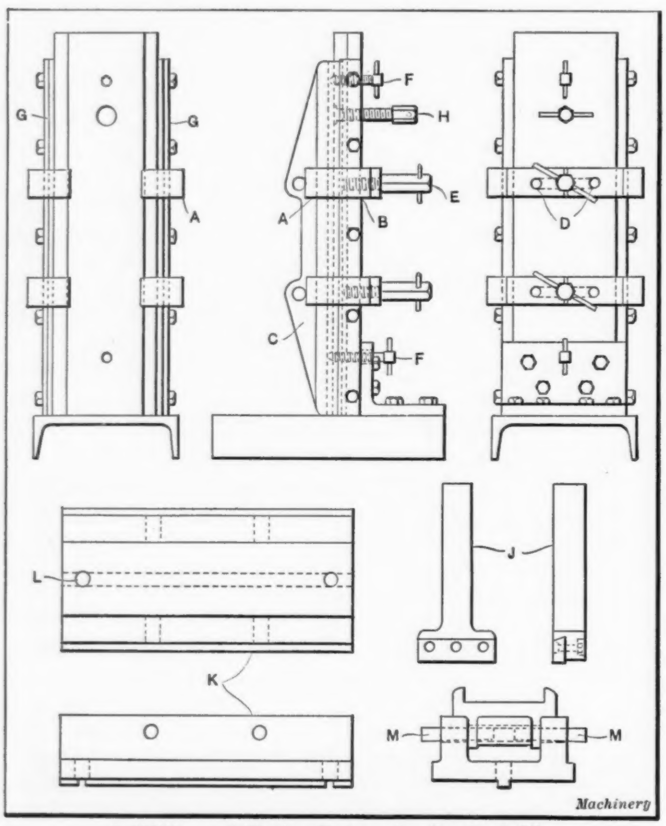
As is also well known, when an engine is being hauled by another engine, it is supposed to be attended by some person

who should have noticed this accident sooner. There was an attendant in this case but he had left his charge and was riding the leading engine. The chains shown in the illustration were put on and the engine hauled into the shop.

Birmingham, Ala. JOHN A. COOK

BABBITTING AND PLANING CROSS-HEAD GIBS

Appliances for babbitting and planing locomotive gibs, which proved very satisfactory in a Western railway shop, are shown in the accompanying illustration. The three upper views show the mandrel or fixture used for babbitting the gibs. The four U-shaped clamps *A* act as guides and hold the gib in place while being babbitted. These clamps engage shallow grooves at *B* which are cut in the back of the mandrel body. When the gib *C* is placed in position for babbitting, as shown, the clamps *A* are pushed in against stops *D* and the screws *E* are tightened. The stops *D* are provided so that the gib will be held at the proper distance from the face of the babbitting fixture. The planed surface on the back of the gib rests against the clamps so that the face of the gib is held parallel with the fixture. Screws *F* hold the gib out against the clamps.



Babbitting and Planing Fixtures for Crosshead Gibs

Before pouring the babbitt, fireclay is placed along the sides to prevent the metal from running out. This is held in place by flanges *G*. When the babbitt has hardened, clamps *A* are loosened and the gib is forced off the mandrel by the screw *H*, the side of the mandrel being slightly tapering so that the gib can be removed easily.

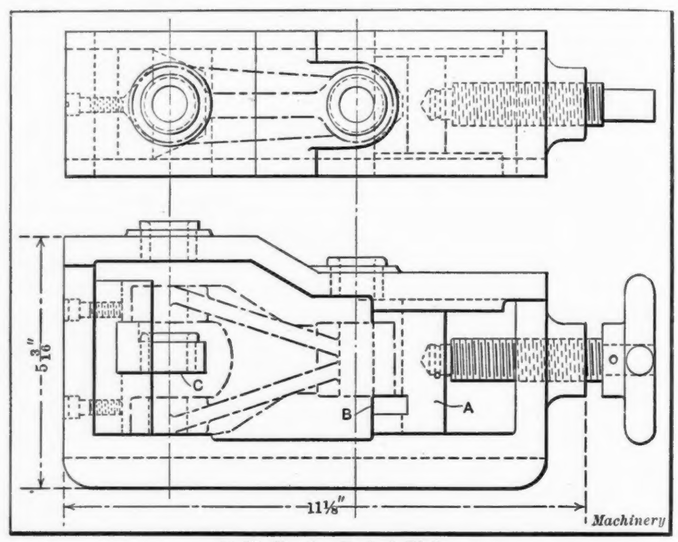
When this mandrel is used, the babbitt is distributed uniformly and the flanges have practically the same thickness, so that it is unnecessary to leave more than 1/16 inch of metal to be planed off. For this reason the mandrel is superior to types generally used, because it reduces the time required for planing.

The planing tool used for finishing the babbitted surface is shown at *J*. The cutter is fastened to the holder with three small round-headed bolts as shown. The width of the cutter is slightly less than the stand rd width of the guide so that a gib can be planed to fit a guide which has become worn. This tool has only a slight amount of clearance on the bottom, as excessive clearance causes it to gouge into the metal. The chuck or fixture used for holding the gibs while they are being planed is shown at *K*. This fixture is held to the planer table by bolts passing through holders *L* and is aligned by a tongue on the bottom. The gib is held by taper pins *M* which are driven in through the flanges at the side, as shown in the end view. The holes in the fixture are a little lower than those in the gib so that the pins will draw the gib down firmly against the top of the chuck. The planer used for this work has a cutting speed of 50 feet per minute and the gibs are planed at the rate of 5 per hour, which is three times the production obtained before these tools were made.

H. T. P.

DRILL JIG FOR FORK LINKS

The drill jig shown herewith was designed for drilling fork links. The form of these links is indicated by dot-and-dash lines in both views. The link has a round boss at one



Drill Jig for Forked Links

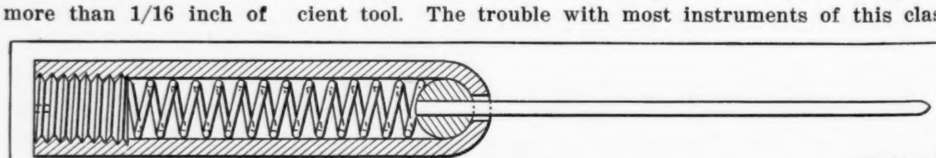
end and rounded forks at the other end. It is accurately held between two V-blocks, one being adjustable and the other stationary. The adjustable V-block *A* is clamped against the work by the star-wheel and screw shown, and it travels between finished ways, thus providing an accurate as well as rapid method of clamping.

These V-blocks have inserted steel plates *B* and *C*. The latter, which is in the stationary V-block, carries a drill bushing for drilling the lower fork, and an upper shoulder on this plate provides a support for the upper fork; thus there are two bushings in alignment for drilling the two ends. The inserted plate *B* in the adjustable block supports the opposite end of the fork link. With this arrangement, a two V-clamping jig is obtained having a three-point support. This drill jig was accurate, rapid and easily operated.

M. W. W.

INDICATOR FOR LOCATING CENTER PUNCH MARKS

An indicator for use in locating center punch marks is shown in the accompanying illustration. This form of indicator can be used on the milling machine, boring mill or drill press, and in the writer's opinion is a particularly efficient tool. The trouble with most instruments of this class



Simple Form of Indicator for locating Center Punch Marks

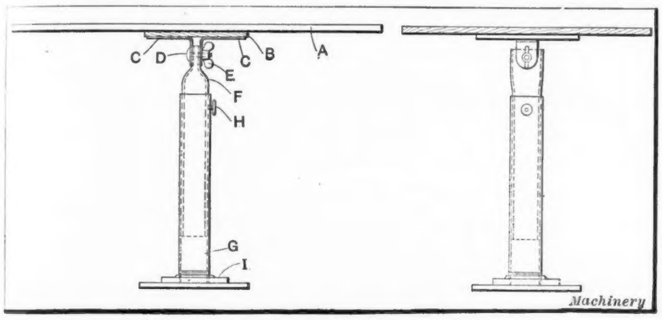
is that the body is not in perfect alignment with the hole; but with the present design, the hole can be lapped and the outside ground in accurate alignment with it.

Milwaukee, Wis. W. BUTZLAFF

AN INEXPENSIVE DRAWING BOARD

The accompanying illustration shows a cheap and satisfactory drawing-board which may be made by almost any one at a small cost. The writer has used a board of this description for some time, and found that it meets all requirements that a more expensive one does. The cost of the complete outfit, including a little work that had to be done outside, was about $5.

Referring to the illustration, *A* is the drawing board. A board *B*, about 15 inches square by 1 inch thick, is secured to the under side of the drawing board by five or six wood screws, in such a way that the grain of the wood in the two boards is at right angles. The plates *C* are made of flat steel, ¼ inch thick by 2½ inches wide. Each of these plates is fastened to the board *B* by means of three wood screws.



Design for an Inexpensive Drawing Board

Holes are drilled in the lower edge of these plates for a ½ inch carriage bolt, the hole in one plate being filed square to receive the head of the bolt *D*. The nut *E* on this bolt is an ordinary wing-nut, which may be obtained in any hardware store.

The support *F* is made of a piece of ordinary 3-inch wrought iron pipe, two feet in length. One end of this pipe is hammered flat so that it will fit between the plates *C*, and the hole is drilled to receive the bolt *D* which holds the table to the support. The lower section of the support *G* is a piece of 3½-inch iron pipe, one end of which is screwed into a standard 3½-inch floor flange *I*. The height of the drawing board can be regulated by adjusting the position of the pipe *F* in pipe *G*, and then screwing up the binding screw *H*. The flange at the end of the support is screwed to a board which should be about 18 or 20 inches square.

It will be readily seen that the design of this drawing board makes it possible to have it set at any desired angle. When the board is not in use, it may be folded down into a vertical position and then pushed over against the wall, where it occupies very little space. The pipe sizes referred to were selected because the outside diameter of the smaller pipe fits nicely in the inside diameter of the larger one. Furthermore, the standard flange used on the larger pipe is heavy enough so that it will hold the table without requiring it to be screwed down to the floor.

Indianapolis, Ind. FRED E. HOSMER

HARDENING DRAFTSMEN'S RULING PENS

During my experience as a draftsman, I have found that even the highest-priced ruling pens do not always have the proper temper to hold a satisfactory point for any length of time. If a pen is too soft, it requires frequent touching up with a hone and as is well known, one or two improper strokes will spoil the result of the most careful effort to obtain a good working point. I use the following treatment for sharpening a pen. First use an ordinary stone of close medium grit. With such a stone and the proper care, a pen can be brought to a very satisfactory point. In order to be sure that the pen is of the desired hardness, I use the following method of hardening: After the pen has been sharpened, the screw is removed and the pen heated with a Bunsen burner or any other gas flame which does not smoke. The flame of an ordinary gas range can be used with satisfactory results. The heating is commenced at the point, care being taken to turn the pen continually in order to have both points evenly heated. The pen is heated slowly until a bright red is obtained about 3/16 inch back from the point.

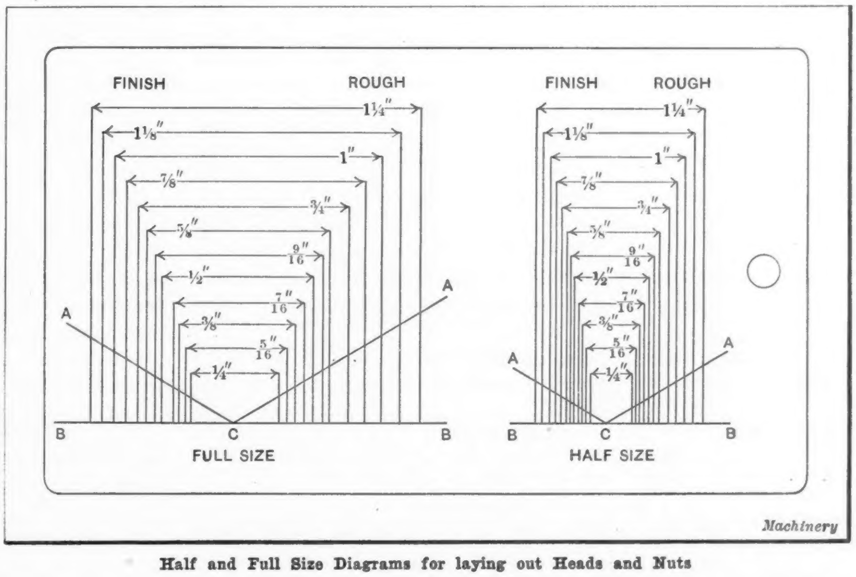
I have found that the best way of quenching the heat is by thrusting the point of the pen into a cake of beeswax to a depth of about ¾ inch. If beeswax is not obtainable, a cake of clear castile soap is almost as good. The pen is left in this position until the wax sets around it and is then allowed to cool for about an hour, after which it can be removed and the wax cleaned off with gasoline or benzine. The point is then brought to a final finish by using a fine oilstone. Of course, the original polish will have been spoiled by this treatment but it may be easily restored by using a piece of crocus cloth.

Fort Wayne, Ind. HOMER R. TALMAGE

LAYING OUT HEXAGONAL HEADS

The accompanying illustration shows a convenient method of laying out hexagonal heads and nuts and the following gives a brief description of the way in which a diagram of this kind is laid out. On *BC* lay off from *C* distances equal to one-half the length across the flats of the heads and nuts most commonly used. The distances on one side of *C* are made to represent the finished sizes and on the other side of *C* the rough sizes. Next, draw lines to form the 30-degree angles *ACB*. Then draw vertical lines from the points laid off on *BC* and connect the top of these verticals on the "finished" side with the tops of the corresponding sizes on the "rough" side. Mark each horizontal line with the diameter of the bolt which the line represents.

It will be evident that with the compass set from the point *C* to the proper point on the line *BC*, a circle can be drawn, around which the plan of the head or nut can be laid out. To draw the side elevation without the plan, the compass is set from the point *C* to the intersection of the proper vertical line with the line *AC*, which represents one-half the distance across the corners. This distance is then laid off from the center line of the bolt. The compass is then set from the proper point on the line *BC* to the intersection with the line *AC*, which represents one-half the side of the head or nut, and this distance is laid off on each side of the center line of the bolt. After these dimensions have been secured, the head or nut can be laid out in the ordinary way.



Half and Full Size Diagrams for laying out Heads and Nuts

I use a piece of bristol board carefully laid off with fine lines. The size of the card is made 2¼ by 4¼ inches. I have found that two charts, one full size and the other half size, are most convenient for my work, but the same principle can be applied to other scales.

Barberton, Ohio. L. E. PARKER

HOW AND WHY

QUESTIONS ON PRACTICAL SUBJECTS OF GENERAL INTEREST

UNIVERSAL TYPE METAL

R. O.—I have heard of a new type metal that can be successfully used for linotype machines, monotypes and for stereotyping, and would like to know the constituents.

A.—You doubtless refer to a new alloy compounded by George R. Wagner, chief linotype machinist of the *New York World.* This alloy is said to work equally well for linotypes, monotypes and stereotypes. It consists of lead 82 per cent, antimony 13 per cent and tin 5 per cent. It is not patented.

HIGH-SPEED STEEL JIG BUSHINGS

C. R.—Is it practicable to use high-speed steel for jig bushings? What are the advantages of high-speed steel for this purpose?

A.—The advantages that would result from the use of high-speed steel for jig bushings would hardly seem sufficient to warrant its use. The high cost of high-speed steel and the difficulty of working it would seem to make its use practically prohibitive. The question is submitted to readers for discussion.

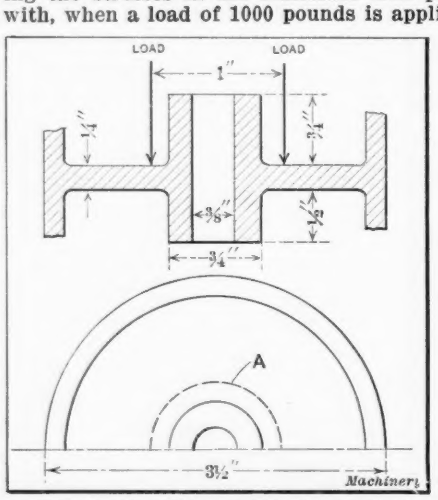
PRESSURE REQUIRED TO UPSET COLD STEEL

W. J. S.—How much pressure is required to upset a 3/8-inch round rod of machinery steel, the head formed to be 5/8 inch diameter?

A.—Experiments made by William Kent and recorded in his "Mechanical Engineers' Pocketbook" indicate that soft steel rivets require a pressure of about 100,000 pounds per square inch for cold heading. If the maximum diameter of the head on the rod is 5/8 inch, its cross-section area is nearly 1/3 square inch; hence, a maximum pressure of 30,000 to 33,000 pounds would be required to form the head.

CALCULATING STRESSES IN MALLEABLE IRON PLATE

W. N. A.—I would like to know how to proceed in calculating the stresses in the malleable iron plate illustrated herewith, when a load of 1000 pounds is applied on the circumference of a circle *A* 1 inch in diameter. Do the bosses 3/4 inch high and 1/2 inch high increase the strength of the plate? Would this be considered a circular plate fixed at the edge, with a concentrated load in the middle and a radius of 1/2 inch? There is a 3/8-inch rod passing through the boss in the middle of the plate and it is possible that the load may create a tendency for the metal to deflect and grip this rod. What I would like to obtain is formulas that will show the deflection and the stresses in the metal in order to be able to determine whether the design is capable of safely supporting the load of 1000 pounds.



Malleable Iron Plate in which it is required to determine the Stresses

Answered by W. L. Cathcart

A.—Only general answers can be given to these questions for several reasons. First, the diaphragm supporting the load is, technically, an unstayed "flat plate," circular and fixed at the edge, and the theoretical analysis of the resistance to bending of such plates is as yet unsatisfactory. Second, the plate in question has a central hole with bosses or stiffening rings, and this modification materially affects the results from such formulas as have been deduced. Third, while both strength and rigidity are required in this design, the plate is only 1/4 inch thick and it is to be made of malleable iron. For castings no thicker than this, the composition and heat-treatment of this material are usually such that it is far from stiff and is capable of much distortion without rupture—that is, it is strong but not rigid. However, the following discussion of the general principles affecting the design will be of service in judging its fitness for the work.

Stresses—General Methods

Formulas for the stresses in unstayed plates can be deduced in several ways. Thus—following Merriman's method—let Fig. 1 represent a circular plate of radius r and thickness t, supported at the edge and carrying a uniform load of p pounds per unit of area of the upper surface. The total load is then

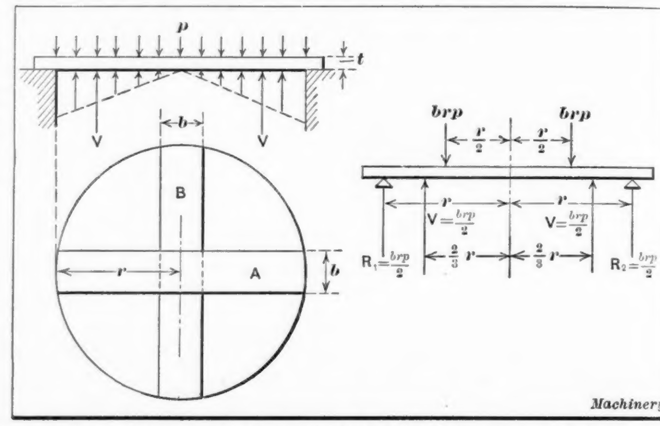


Fig. 1. Diagram showing Method of determining Stresses in Plate

$\pi r^2 p$ and the length of the support of this load is $2\pi r$. Hence, the reaction of that support, per unit of length, is $\frac{\pi r^2 p}{2\pi r} = \frac{rp}{2}$.

Now, take an elementary strip A of width b, whose center line is a diameter. The area of this strip is $2br$, the total load on it is $2brp$, and the reaction at each end is $\frac{brp}{2}$ or brp for both ends. Since, to support a load $2brp$, only reactions of brp are thus available, there must be, for equilibrium, upward resisting shearing forces at the sides of the strip equal to brp, the remainder of the load.

As the load is uniformly distributed, the intensity of these shearing forces is zero at the middle of the beam and increases at a uniform rate to the supports. Hence, their resultant V acts at a distance $\frac{2r}{3}$ from the middle or through the center of gravity of the triangle which they form. The upward and downward system of forces thus constituted is shown at the right-hand side of Fig. 1. The maximum bending moment, which is at the middle of the beam, is $M = \frac{br^2p}{3}$.

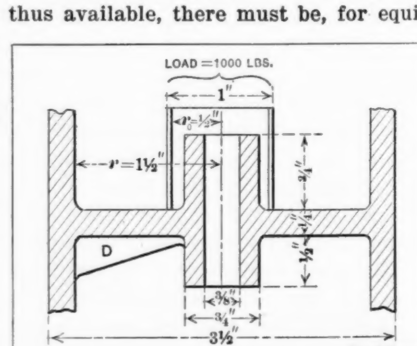


Fig. 2. Cross-section through the Plate

The corresponding stress is:

$$S_1 = \frac{Mc}{I} = \frac{6M}{bt^2} = 2p\frac{r^2}{t^2} \qquad (1)$$

This is, however, an apparent and not a true stress, since allowance must be made for the modifying effect of similar strips crossing A at the center. Thus, for a strip B at right angles to A, we have a similar stress S_2, equal to S_1, but per-

pendicular to it. Rupture generally occurs on the lower or tensile side of the plate. For that side, the true maximum tensile stress is then $T = S_1 - eS_2$, in which e is the factor of lateral contraction. The mean value of e for iron and steel is 1/3. Substituting this value and those of S_1 and S_2, we have for a supported plate:

$$T = \frac{4\,r^2 p}{3\,t^2} \tag{2}$$

For a plate fixed at the edge, the determination of T is much more complex and cannot be given here. In such plates, this stress is, however, about three-fourths of that given by Equation (2) or:

$$T = \frac{r^2 p}{t^2} \tag{3}$$

Stresses due to Concentrated Loads

The method of analysis given above is general and can be modified to determine the stresses due to concentrated loads. For the case in question, as shown by Fig. 2, a load of 1000 pounds is concentrated concentrically on a circumference whose diameter is 1 inch. For these conditions, with a supported plate having no central hole or bosses, the formula becomes:

$$T = \left(\frac{4}{3}\,\text{hyp. log.}\,\frac{r}{r_0} + 1\right)\frac{P}{\pi\,t^2}$$

where T = safe working tensile stress;
r = radius of plate;
r_0 = radius of load circle;
P = load in pounds;
t = thickness of plate in inches.

In this case, the radius r is 1½ inch, $r_0 = ½$ inch, and $r \div r_0 = 3$, the hyperbolic logarithm of which is 1.1. Substituting and transforming:

$$T = 0.77\,\frac{P}{t^2} \tag{4}$$

for a supported plate. Taking three-fourths of this result, as in Equation (3), we have, for a plate fixed at the edge:

$$T = 0.58\,\frac{P}{t^2} \tag{5}$$

For malleable iron in tension, average values are: ultimate strength, 40,000 pounds per square inch and elastic limit, 18,000 pounds. Taking 5 as a factor of safety, the safe working stress $T = \frac{18{,}000}{5} = 3600$ pounds per square inch. Substituting this value and $P = 1000$ in Equation (5) we have:

$$t = 0.4 \text{ inch} \tag{6}$$

While this theoretical formula gives, in general, a greater thickness than is usual in practice, this discrepancy diminishes with thin castings, in which several elements combine to lessen the margin of safety.

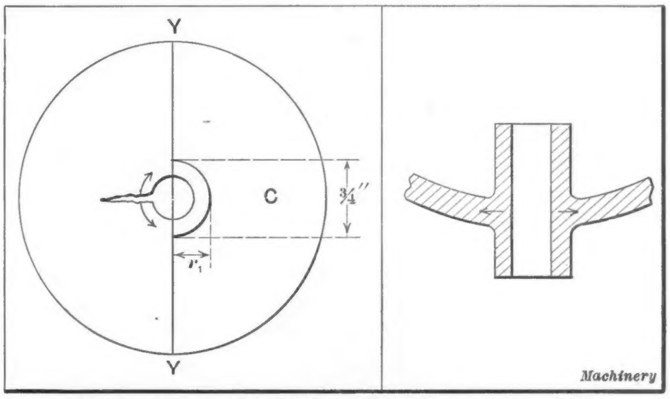
Central Hole and Bosses

A central hole, not reinforced by bosses, of course weakens the plate. This is shown conclusively by Formulas (1) and (2). Owing to the interaction of the stresses through lateral contraction, the relation between the stress in strip A, before it is crossed by strip B, and its stress after that crossing is as 2 to 4/3. That is, the crossing and consequent support lessen the stress in A, not only at the center but throughout its whole length. Since the hole is relatively small, the bosses make up, partially or wholly, for this loss of strength. In the first place, when a plate thus centrally perforated is tested to destruction, as shown at the left-hand side of Fig. 3, there is a circumferential stress at the edge of the hole, so that the plate fails by radial tearing. Therefore, any thickening of this edge will provide more metal and lessen the stress per square inch. It is evident that the wider the boss—that is, the greater the radius r_1—the better, in this respect. Again, take any elementary diametral strip, ¾ inch wide, as C in Fig. 3. At the middle on the diameter YY, there is $3/16 \times 2 \times 1½ = 9/16$ square inch of metal, as compared with $¾ \times ¼ = 3/16$ square inch at any other part of the strip. Finally, the location of this additional metal also increases the strength of the plate, since it increases the depth of the imaginary beam C at the bosses, and the strength of a beam of rectangular cross-section varies directly as the square of its depth.

As to the possibility of the bore contracting and gripping the rod passing through it: in order to bend, the plate must stretch—always at the lower surface, and, with ductile metal, on the upper side as well. This stretching will produce a radial tensile stress in the boss where it joins the plate, as shown in Fig. 4, and this stress will tend to increase, not decrease, the diameter of the bore. For such metal and diameters, a calculation of this stress would have no value. The difference between seizing and a close working fit would be measured in thousandths of an inch, and such minute dimensions can be computed effectively only for material like gun-steel whose composition, heat-treatment, and behavior under strain are known in detail.

Summing Up

Owing to reinforcement by the bosses, the plate is at least as strong as if it had no central hole. Then, considering it as a solid plate, Formula (6) shows that, as designed, it has



Figs. 3 and 4. Possible Results of Excessive Stresses on Plate

but 60 per cent of the required thickness. On the other hand, for heavier castings, this formula gives thicknesses which are too great; also, a low working stress has been used in the calculations; and finally, the reinforcement of the bosses may stiffen the plate sufficiently to make the designed thickness enough for strength.

The difficulty in deciding is due mainly to the thinness of the casting. In heavier plates, malleability makes the iron capable of resisting shock without material bending or breaking, but in thin castings this characteristic is usually carried to an extreme, so that the metal becomes soft and pliable. Further, slight changes in the proportion of silicon or in the treatment during melting and annealing may, with these thin castings, make a marked difference in their physical characteristics. If maximum lightness is desired, a practical way to secure it would be to make a trial casting with a moderately thick plate and the bosses as wide as possible. Then, test the plate, and, if no noticeable deflection appears, machine off a thin layer of metal, and test it again. The loss of the "skin" in this process will not affect the strength of the plate to a greater extent than the removal of any other layer of equal thickness. As a last resort, if the design will permit, the plate may be ribbed underneath, as shown at D in Fig. 2.

Questions and answers of general interest only are published. All inquiries must be accompanied with name and address to receive attention. Compliance will enable some answers to be made by mail if not deemed suitable for publication.—Editor.

* * *

The application of centrifugal force to clothes and metal chips for the extraction of water and oil, is well known, but not so well known is the use for pressing certain materials such as cork. It has been found impossible to compress scrap cork into cylindrical shapes, because of the friction of the material on the sides of the container. In a centrifugal apparatus, however, compression to any form is readily accomplished. Long cylinders of granulated cork are formed without trouble; each particle of cork is impelled outward by the centrifugal force and neither external nor internal friction of the material hinders the compressive action.

NEW MACHINERY AND TOOLS

THE COMPLETE MONTHLY RECORD OF NEW DESIGNS AND IMPROVEMENTS IN AMERICAN METAL-WORKING MACHINERY AND TOOLS

GRIDLEY AUTOMATIC MULTIPLE-SPINDLE DRILL

In developing the Gridley automatic multiple-spindle drill, the intention of the Windsor Machine Co., Windsor, Vt., was to produce a machine applicable to the rapid drilling of parts which are difficult to handle on other types of machines. Tests which have been conducted on these machines in the builder's factory show that there is also a marked saving of floor space in addition to a reduction of time, as compared with other methods that were previously considered satisfactory.

The Gridley automatic multiple-spindle drill is, in reality, a vertical turret machine adapted for drilling, reaming, counterboring and facing a variety of classes of work. The design differs radically from other drilling machines in that the spindles are adjustable both radially and circumferentially, thus enabling all spindles to be located for operation at a common center or at different points, as may be required for various classes of work. Holes can be drilled cutting into each other or as far apart as the capacity of the machine will allow. The spindles are individually adjustable, so that any tool may be placed in position to work to

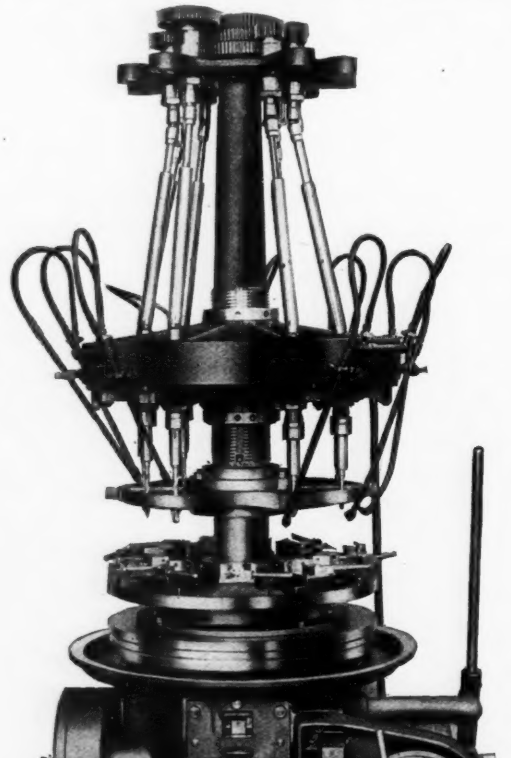


Fig. 1. Front View of Gridley Automatic Multiple-spindle Drill

any depth, regardless of the position of other tools. As all spindles are supported from the center column and distributed around an entire circle, the thrust on one side is balanced by the thrust at the opposite side. This relieves the center column of undue strain, and any tendency for heavy feeds to spring the machine is practically eliminated.

Front and side views of the machine are shown in Figs. 1 and 3, and Figs. 2 and 4 show closer views of the work-table and guide bushing holders that are employed on these machines. It will be seen that the work is held in chucks or suitable fixtures mounted on a rotating work-table which revolves around the center column. As it is necessary to have one position of the work-table idle for the purpose of

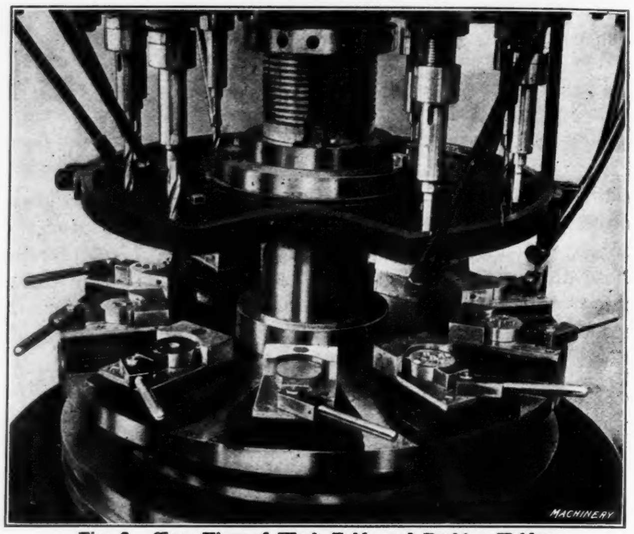


Fig. 2. Close View of Work Table and Bushing Holders shown in Fig. 1.

removing the finished piece and inserting a fresh blank, one more chuck or fixture is used than the number of spindles on the machine. This idle or loading position is clearly shown at the center in Figs. 1 and 2, where it will be seen that the blank has been placed in the fixture ready for the first indexing. In this case the machine is operating on a small threading die. The die shown to the right of the loading position is finished and will be removed when the table has advanced one point; a fresh blank will then be mounted in place ready for the cycle of operations that is to be performed on it.

The necessary number of spindles can be built into the machine to meet the requirements of the class of work to be operated upon, from five to nine spindles being the usual number. In order to divide the time of the successive operations as uniformly as possible, it is sometimes advisable to use two or more spindles working in the same hole, each of which machines a portion of the whole depth. The table indexes one point each time it is lowered from the tools, and one finished piece of work is produced each time the table moves upward. The time taken to produce the piece is that of the longest operation plus the idle time required to bring the work to the tools in the next position. Each individual spindle can be geared to the correct speed for the tool which it carries, and in cases where extra heavy service is required of one or more tools, special spindles of heavy construction can be used for these operations.

In drilling a layout of holes, to insure accurate spacing between them either of two methods of holding the drill bushings may be used. When only a few of the drills or other tools need guide bushings, holders carried by adjustable vertical and horizontal arms are employed. This arrangement is clearly shown in Fig. 4. In cases where a large number of holes are to be drilled, a plate is used which extends around the central column of the machine, the plate being located between the work-table and the spindles. There are large holes in this plate, one of which is in line with each of the spindles. Adjustable bushing holders which can be quickly set in the required position are clamped over these holes. In changing from one job to another it is only necessary to change the bushing holders, and on some classes of work it is necessary to change but one or two of the holders. This is the arrangement which is shown on the machine illustrated in Figs. 1 and 2. For drills and some other tools, hardened steel bushings are used. Other tools which have only short

flutes work more advantageously in bronze bushings which locate the tools properly and prevent all tendency to chatter when the machine is working at high speed.

Little difficulty is experienced in setting up the machine for operation on different pieces of work. The tools are inserted in their holders and a perfect sample of the piece to be machined—finished to the required size and form—is placed in the chuck or fixture. The table is next advanced by hand until the piece is located at the point under the first spindle, where the sequence of operations commences. The spindle is then swung into exactly the required position and the guide bushing fastened in place. After raising the table to its highest cutting point, the spindle is adjusted vertically in order that the hole may be machined to the proper depth. This operation is repeated for each spindle until all of the tools have been properly located.

By referring to the table of spindle speeds, the proper gears are next placed on the center shaft and spindles, and the

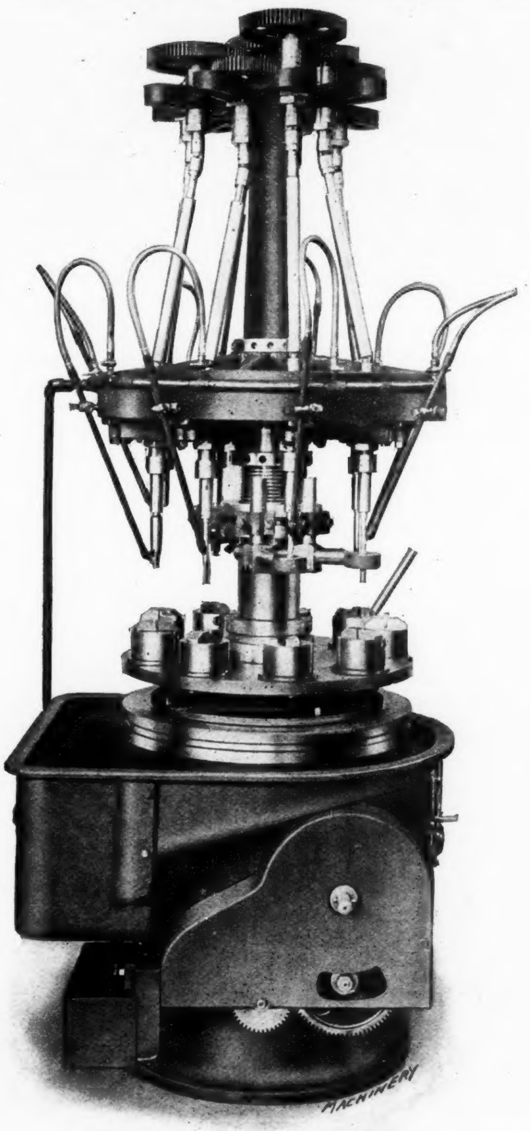


Fig. 3. Side View of Gridley Automatic Multiple-spindle Drill

gears brought into mesh. The change gears which give the different speeds for the vertical center shaft are located in the case at the lower left-hand side of the machine in Fig. 1. The change gears giving the different feeds to the table are also located in this case. After the proper speed and feed gears are in place, the table should be brought up to the position where the first tool of the set nearly touches the work. At this point, a cam-pin on the operating disk should be set to throw out the fast feed clutch; this will allow the feed gears to drive the table at the proper feed. After the table has reached its highest point and the tools have finished cutting, the second cam-pin on the operating disk should be set to throw in the fast feed clutch. The machine is then ready to start working.

The Gridley automatic multiple-spindle drill is equipped with a gear-driven oil pump, an oil tank and a separate oil supply pipe with an adjustable nozzle to deliver lubricant to each cutting tool. The arrangement of the piping, flexible tube connections and nozzles will be readily understood by

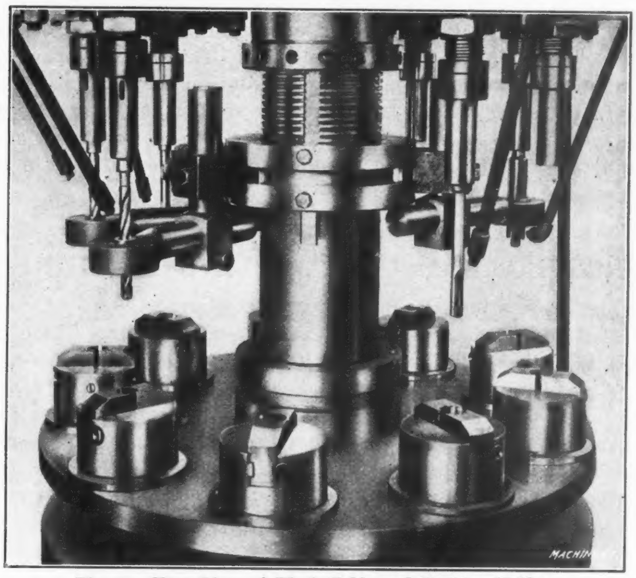
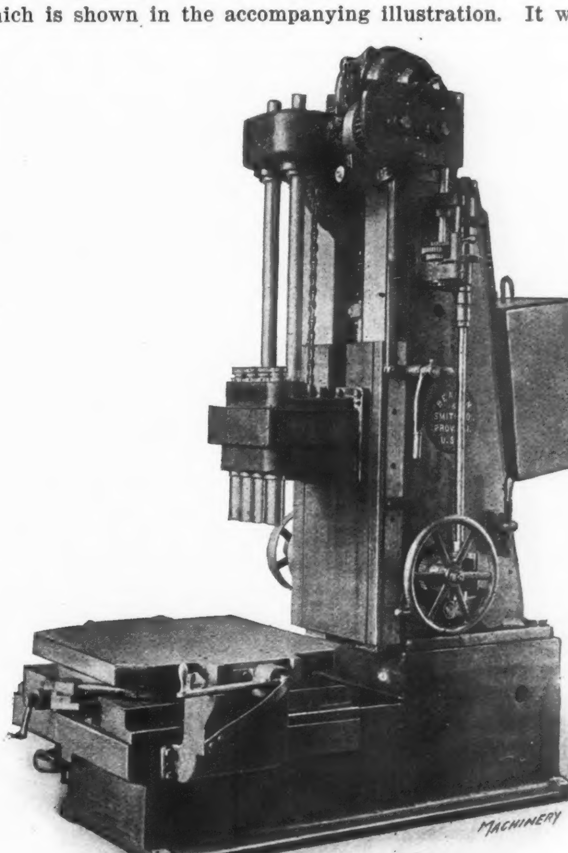


Fig. 4. Close View of Work Table and Bushing Holders shown in Fig. 3

referring to the illustrations. The machine is equipped with a self draining drip tank and strainer of liberal size. There is ample room for removing chips without requiring the oil guards to be disturbed in any way.

BEAMAN & SMITH BORING MACHINE

The Beaman & Smith Co., Providence, R. I., has recently added to its line a vertical spindle cylinder boring machine which is shown in the accompanying illustration. It will be



Beaman & Smith Four-spindle Cylinder Boring Machine

seen that this machine has four spindles, but these machines are built with either three, four or six spindles. The spindles run in taper bearings lined with hard bronze, means being provided to compensate for wear. The head is carried by a counterbalanced saddle which has a vertical movement of

29½ inches. The spindles have automatic feed and automatic stops are provided to trip the feed at the required point; quick power movements are also provided.

The work is set up on a revolving work table 36 inches square. There are two index points, and the finished work, which is held in suitable fixtures, can be removed from the idle position while the boring operation is being performed on a casting in the operating position. This feature provides for keeping the machine in almost continual operation. The table is supported on ball bearings so that it is easily revolved. The machine is driven by a 7½ horsepower motor and transmission of power to the spindles is through a train of spiral and spur gears.

ROWBOTTOM CAM MILLING MACHINE

The Rowbottom Machine Co., Waterbury, Conn., has recently perfected a universal cam milling machine which is adapted for cutting all types of cams. It is said to be the first cam cutting machine which has been designed with the cutter head carried on a vertical slide. Several important advantages are secured through this construction, among which the following may be mentioned: The roller which governs the movement of the cutter head is always kept in

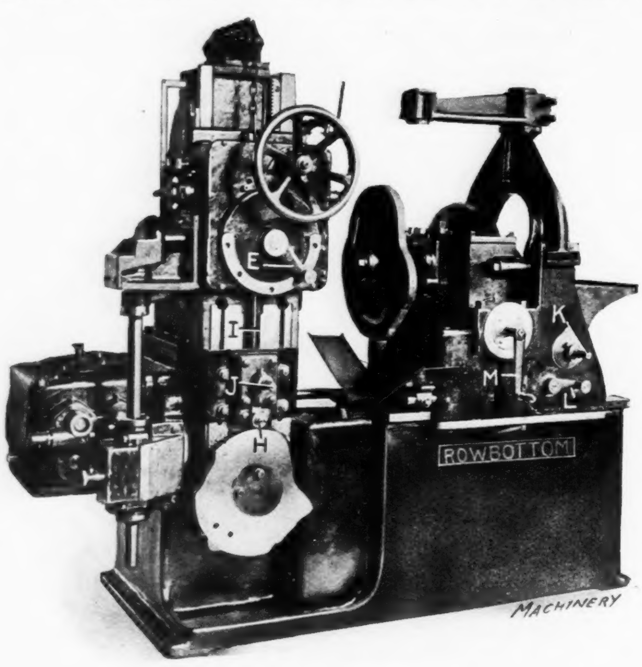


Fig. 1. Front View of Rowbottom Universal Cam Milling Machine

contact with the former cam through the action of gravity. When the cutter head works in a horizontal plane there is frequently a tendency for the cutter to creep or "flinch" away from the work, especially when difficult sections of the cam are being machined. Another advantage of the vertical cutter slide is that the chips and dust produced in cutting drop away from the work as fast as they accumulate.

Referring to Figs. 1, 2 and 3, it will be seen that the cam to be milled is held on a mandrel carried by the work head. This head may be supported with the mandrel in either a horizontal or vertical position as shown in Figs. 1 and 3. For milling plate or face cams, the mandrel is held horizontally, and for machining cylindrical cams the vertical position of the mandrel is employed. When using the mandrel in the vertical position, the outboard support is used; this support may be removed from the machine when it is not needed. The change from the horizontal to the vertical position is effected by first removing the support under the faceplate and then swinging the head down. The bolts which were formerly employed to secure the head in the horizontal position are now located in the proper position to clamp the head vertically. These bolts enter holes in the work head slide at the place where the faceplate support was formerly secured.

The machine is driven through a single pulley which is belted to a two-speed countershaft. The single driving pulley is equipped with a Johnson clutch, which is operated by the lever *A* located on top of the gear-box in Fig. 4. This lever is within easy reach of the operating position and provides for instantly starting or stopping the machine. The small lever *B* operates a sliding clutch between two bevel gears, and by throwing this lever to the forward or reverse station, the clutch is engaged to drive the machine right- or left-hand

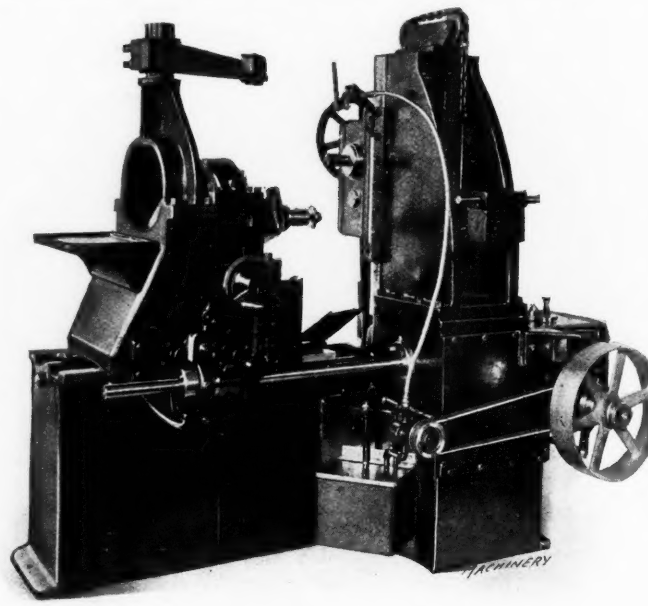


Fig. 2. Opposite Side of Machine shown in Fig. 1

as desired. There is also a neutral station for the lever *B*, in which position the clutch is out of engagement with both gears. It will also be seen that there is a lever *C* mounted at the rear of the gear-box which is provided with four stations. This lever operates a cone of gears in the gear-box to give either of four speed changes, and as the driving pulley is belted to a two-speed countershaft, there is a total of eight changes of speed available. Each of the speeds for the

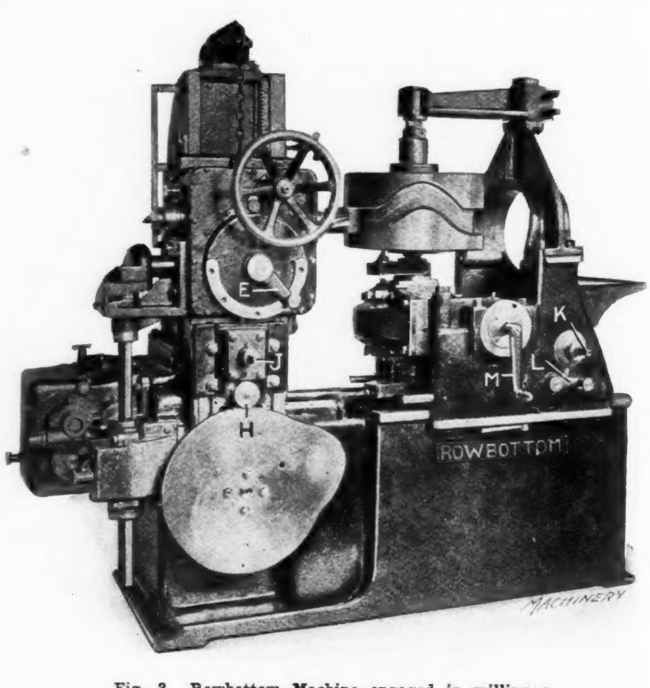


Fig. 3. Rowbottom Machine engaged in milling a Cylindrical Cam

former spindle and work spindle obtained in this way are necessarily the same in order to maintain the proper relation between the cutter and the work.

The last of the four operating levers in Fig. 4 is shown at *D*; it will be seen that this lever has three stations, the central one of which is the neutral position, while the other two stations provide forward and reverse rotation for the cutter spindle. An ingenious mechanism is used to obtain four

speed changes for the cutter spindle, and as in the case of the former and work spindles, there are eight available cutter speeds, owing to the drive being taken from a two-speed countershaft. An important feature of this cutter spindle speed changing device is that all changes are obtained through a single lever, which is shown at E in Figs. 1 and 3. These changes of speed are obtained through a sliding gear transmission which is shown in detail in Fig. 5. Referring to this

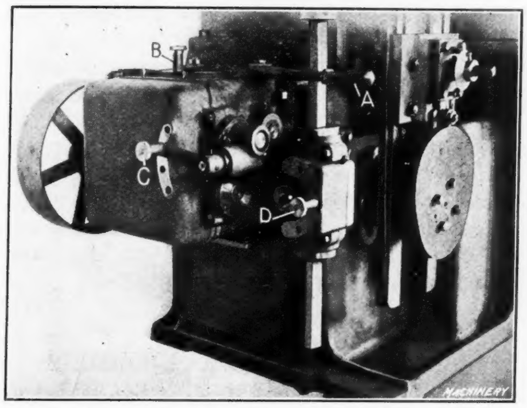


Fig. 4. Close View of Gear-box on Rowbottom Cam Miller

illustration, it will be seen that there are four pairs of gears; the four gears carried on the cutter spindle are fixed longitudinally and the same applies to the gear at either end of the driving shaft. Each of the driving gears F and G are splined to the shaft and provided with positive clutches, by means of which they may be connected to the gears at either end of the driving shaft. The longitudinal movement of these gears is controlled by fingers carried by two cam operated slides, and either of the gears F or G may be engaged directly with a gear on the cutter spindle or the clutch on either of these gears may be engaged with one of the gears at the end of the driving shaft which is always in mesh with the spindle gear. In any case, there is one of the four pairs of gears in mesh which provides for transmitting the motion to the cutter spindle at the required rotative speed.

It was mentioned that the movement of the driving gears F and G in Fig. 5 was controlled by fingers carried by a cam operated slide. The arrangement of this cam mechanism is shown in Fig. 6, the cam being operated by the lever E which is also shown in position on the machine in Figs. 1 and 3. The slides which move the gears F and G along the driving shaft are operated by the rollers N and O, Fig. 6, which run in a cam groove shown in this illustration. When the lever E is in the central station, both clutches and gears are disengaged, and consequently there is no rotation of the cutter spindle. By moving the lever E to the first station to the left

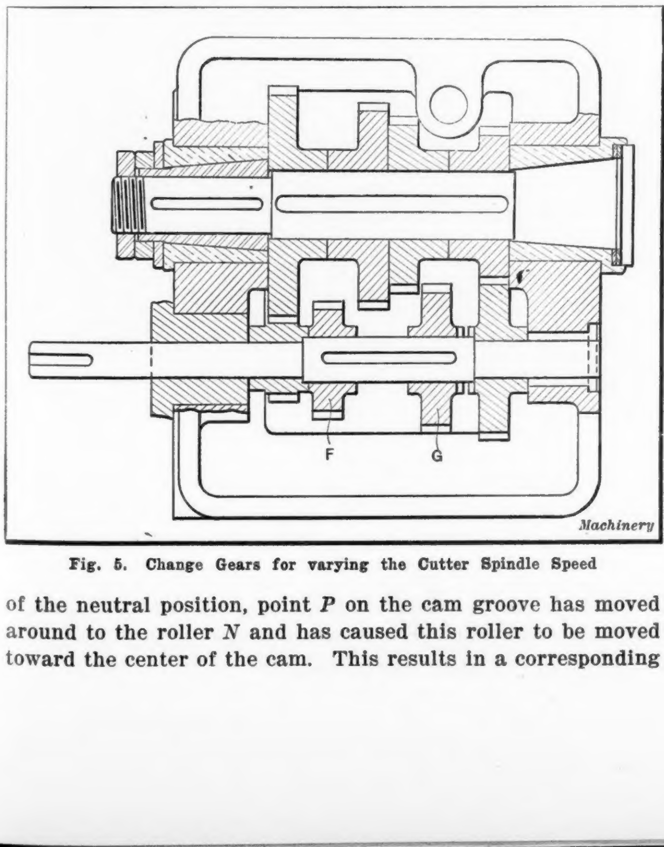


Fig. 5. Change Gears for varying the Cutter Spindle Speed

of the neutral position, point P on the cam groove has moved around to the roller N and has caused this roller to be moved toward the center of the cam. This results in a corresponding movement of the slide, which, in turn, throws the gear F into mesh with the gear on the cutter spindle. By moving the lever to the second station to the left of neutral, the point R on the cam groove has reached the roller N; this causes the roller N to be moved away from the center of the cam, and, in turn, the fingers on the slide move the gear F so that the clutch member on this gear engages the gear to the left of it, thus making the left-hand gear on the shaft the driving gear. By moving the lever E two stations to the right of the neutral position, the points Q and R in the cam groove are successively brought up to the roller O and cause the gear G to drive two of the gears on the cutter spindle as previously described for the gear F. In addition to the eight changes of speed obtainable through the gearing, an independent geared speeder is provided for use when small sized cams are being milled.

One of the features of this cam milling machine is that flat former plates are used for all types of cams. When the rise of the cam to be milled is relatively steep, it is advisable to use a former plate which is double the size of the work, if such a proceeding is practicable. The former plate is bolted to the end of the former spindle, and as the plate revolves it pushes the entire cutter slide up through its action upon the roller H shown in Figs. 1 and 3. This roller is mounted on an auxiliary slide which is connected to the main cutter slide by a screw adjustment, shown at I in Fig. 1. This screw adjustment is operated by a crank on the end of the squared shaft J to move the cutter slide on the bearings of the column, thus providing for operating the cutter at any desired distance from the center of rotation of the work-spindle. A scale is provided on the column which shows the distance that the cutter has been set from the center of the work-spindle. The adjustment for the depth to which the cutter enters the work is made on the work slide, the squared shaft K being provided for this purpose. This shaft has a micrometer collar so that when the work is brought up so that it just touches the cutter, this collar may be set at zero, after which the work is fed to the cutter to provide for milling to the required depth.

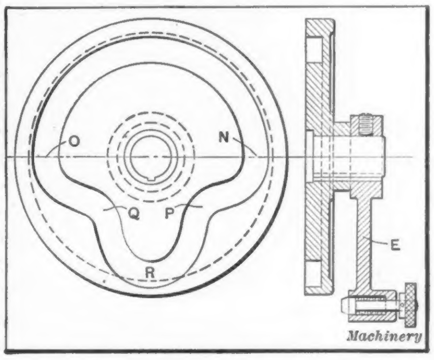


Fig. 6. Cam Mechanism for changing the Cutter Speed Gears

A convenient feature of this machine is the provision that is made for changing the relative position of the cutter and cam blank that is being milled. Suppose it was desired to alter the position of the cam groove slightly from that arbitrarily fixed by the location of the driving pin which passes through the work into the faceplate. The first step would be to move the lever L, Figs. 1 and 3, to the neutral position, which disengages the clutch that drives the work-spindle. The lever M is next turned through 1/12 of a revolution, after which the clutch is reengaged by the lever L. This clutch has 12 teeth and the result of this operation has been to re-mesh the clutch one tooth ahead of its former position. By means of the gearing in the work head, the resultant movement of the work relative to the cutter amounts to ¼ degree. Power is transmitted to the work head by means of the horizontal shaft shown at the back of the machine in Fig. 2, and horizontal movement of the head on the ways of the machine is obtained through a horizontal screw. By reversing the direction of rotation by means of the lever B in Fig. 4, the work head may be traversed either toward or away from the cutter. The total length of a cylindrical cam that can be cut is represented by the traverse of the work head.

No lubricant is used on the cutter, but a strong blast of air is employed which serves to remove all chips and keep the cutter cool. This air blast is found to be an excellent substitute for lubricant even when the machine is engaged in cutting tool steel cams. However, there is a pump provided on the

machine and lubricant may be used if desired. This machine is very rapid in operation and is found satisfactory for milling practically any type of cam which is of a size within its range. The capacity is for face cams up to 28 inches in diameter, box cams up to 32 inches in diameter, and cylindrical cams up to 24 inches in diameter, with an 11-inch throw. The weight of the machine is 4500 pounds.

PORTER-CABLE LATHE

Quite a departure from standard practice has been made in the design of the lathe illustrated herewith, which is a recent product of the Porter-Cable Machine Co., Syracuse, N. Y. This machine is primarily designed for manufacturing operations on short pieces having a maximum length up to 12 inches. The machine gives excellent results when operating on work held between centers and is also well suited for a variety of small chuck and faceplate work. In developing this lathe, the idea was to produce a machine capable of handling the classes of work referred to, and at the same time save the purchaser from paying for a machine of larger size than he requires and one which occupies an unnecessary amount of floor space. This result is made possible by the application of an overhanging tailstock which travels on its own dovetail ways at the rear of the bed. This arrangement enables the carriage to be constructed on the plan of a milling machine table. The bed is 32 inches long and as it is not open at the top it is exceptionally rigid. The ways are protected at all times by the chip guards fastened at each end of the carriage, the front guard passing under the headstock.

Porter-Cable Lathe for Short Work on Centers or Small Chuck Work

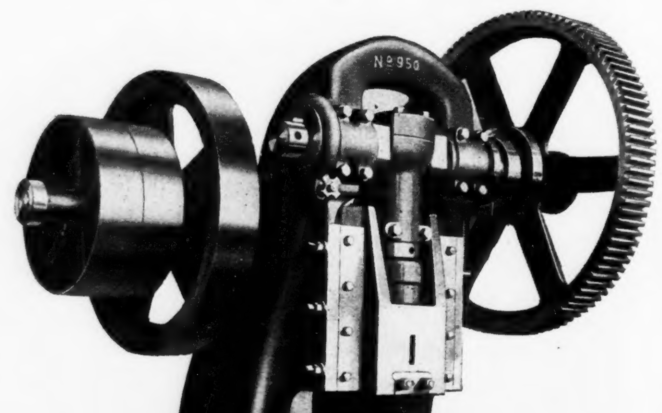
Both hand and power feed are obtained by means of a rack and pinion located nearly under the center of the carriage. This pinion is integral with a vertical shaft extending down through the bed, where it is connected through suitable gearing and a positive clutch to the regular change gears at the end of the lathe. Threads from 5 pitch down to the finest may be accurately cut. The feed in either direction can be automatically disengaged by the adjustable trip located underneath the carriage, and when thrown out the carriage may be run along by hand to the full limit of its travel without removing any stops.

The spindle is exceptionally heavy and can be furnished for use with a plain center or a set of draw-in collets for handling bar stock up to ⅝ inch diameter. The drive, which is of ample power, is obtained by a bevel gear and small pinion running at a high rate of speed, thereby permitting the use of a light belt. The gears are entirely enclosed, and as they run in oil the drive is very smooth and quiet. When it is necessary to change the spindle speed, this can be done in a moment by changing the driving pulley, the difference in the length of the belt being taken care of by the compensating countershaft. A valuable feature of this lathe is that the carriage can be run back underneath the tailstock, thus making it unnecessary to have the tail spindle extended when taking cuts near the dead center or for short work. This is also found to be a convenient feature for such operations as filing, polishing and taking measurements, as the carriage may be run entirely out of the way without disturbing the tailstock. Means are provided for adjusting the headstock sidewise in order to keep the centers in proper alignment. A reservoir for cutting lubricant is located in the upper part of the base and a cupboard in the lower part of the base is provided for keeping tools, gears, etc. For work within its range, the efficiency of this machine is equal to that of a 14- or 16-inch lathe, and it costs considerably less and also effects a saving in floor space.

AMS OPEN-BACK PRESSES

A new line of open-back presses has recently been placed on the market by the Max Ams Machine Co., Mount Vernon, N. Y. The distinguishing feature of these machines is their rugged design which is said to give them the strength of straight sided presses of corresponding size; in addition, strip stock can be fed from the side. Large sheets can be passed through the opening in the back of the press.

The frame is symmetrical in form and adequately ribbed to secure the required rigidity. The die space and the opening in the back of the press are of large size. The bearings of the slide are of ample proportions. A cap clamp, flange or dovetail for holding the punches may be provided according to the requirements of different users. The shaft is machined from a hammered steel forging and adequate provision is made for lubricating the bearings. These presses are particularly adapted for use in the manufacture of the parts of typewriters, locks, hinges and various other hardware and sheet metal specialties. This line of presses comprises eight different sizes and the patterns have been constructed in such a way that the design can be modified to suit the requirements of the different classes of work on which the machines will be used.



Ams Open-back Press

THE PROVIDENCE SHAPER

It has always been conceded that for accurate planing of machine parts having long surfaces that must be square or parallel with each other, it is necessary to use a standard planer. But on many jobs the planer is too slow and cumbersome to give very satisfactory results, owing to its inability to reverse on short strokes when operating at high speed. The slow adjustment of the cross-rail and head also reduces the efficiency of a planer when operating on such work. It has been generally recognized that for efficiency of operation on work that comes within its range, the shaper is a most desirable tool because its mechanism and all adjustments are conveniently controlled.

In designing the Providence shaper, the endeavor of the Providence Engineering Works, Providence, R. I., has been to combine the features of both the planer and shaper in a single machine; at the same time many special features have been added which increase the efficiency with which it operates. It will be evident from the illustrations that the design follows the general lines of an open-side planer, but the cross-rail is supported from the column at the left-hand end instead of at the right, which is the usual practice with open-side planers. Located in this position, the column is not in the operator's way. The open-side feature enables the machine to work on pieces that are too long to go through the housing of an ordinary planer, and also makes both ends of the work visible all of the time, so that the cutting tool can be set to a line at the beginning of the stroke instead of at the end of it. All of the operating handles for controlling the changes of feed and speed are within reach of the operator without requiring him to leave his position at the front of the machine. The head has horizontal, vertical and angular feeds secured through a sliding pinion that may be mounted on either the horizontal or vertical feed-screw, the pinion meshing with the gear on the main feed shaft. The usual type of stroke adjustment is employed.

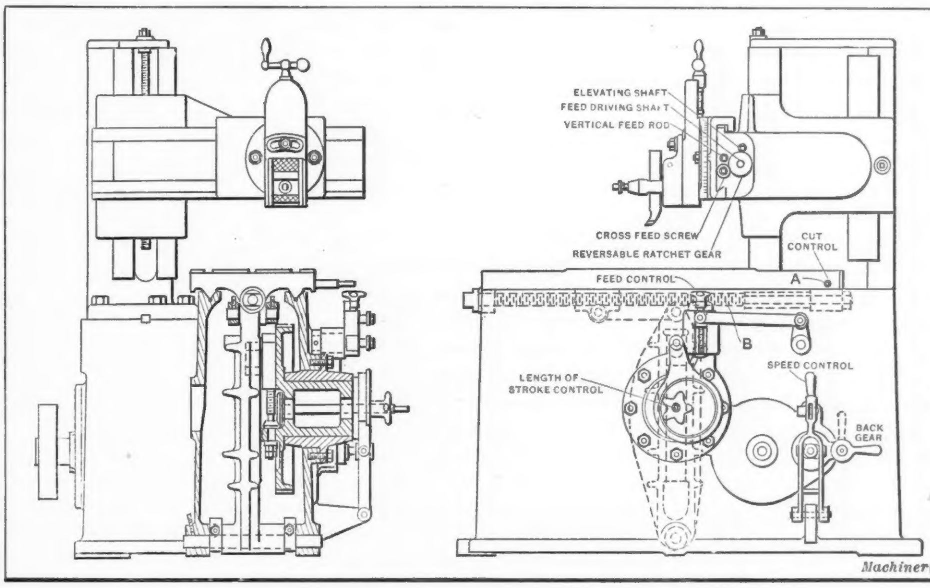


Fig. 3. Design of Machine showing Arrangement of Feed Mechanism

The frame and base of the machine are of box section, and the column is of similar construction. The cross-rail is counterbalanced by a weight located inside the column. Special attention is called to the design of the bearing of the cross-rail on the column. This bearing is longer than it is

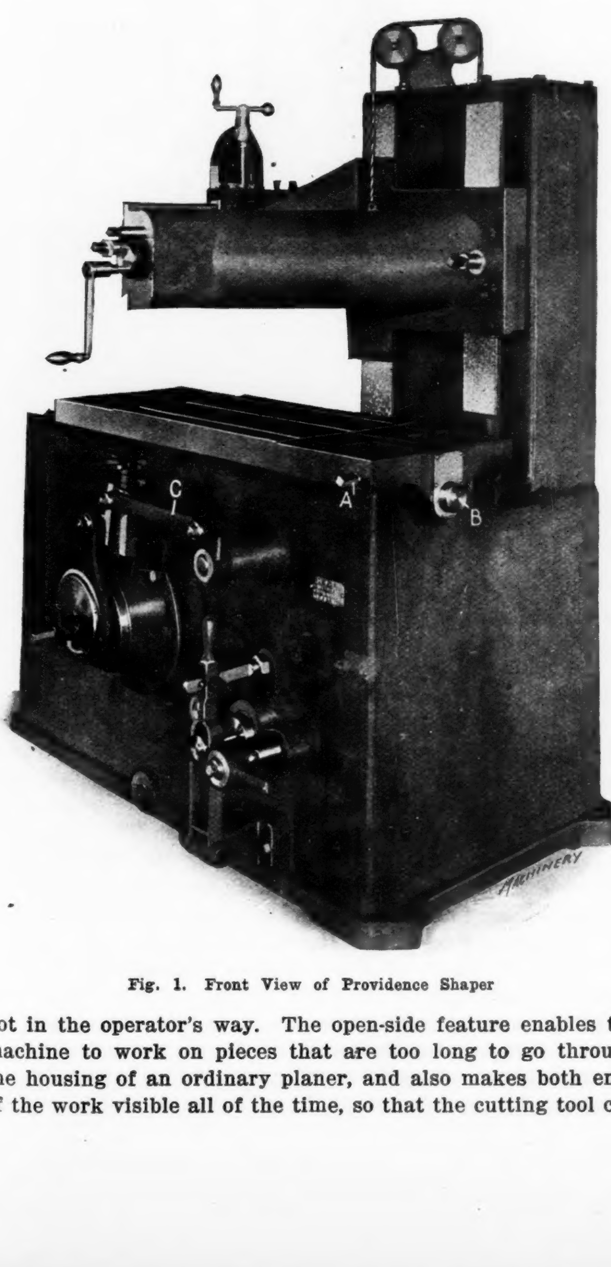


Fig. 1. Front View of Providence Shaper

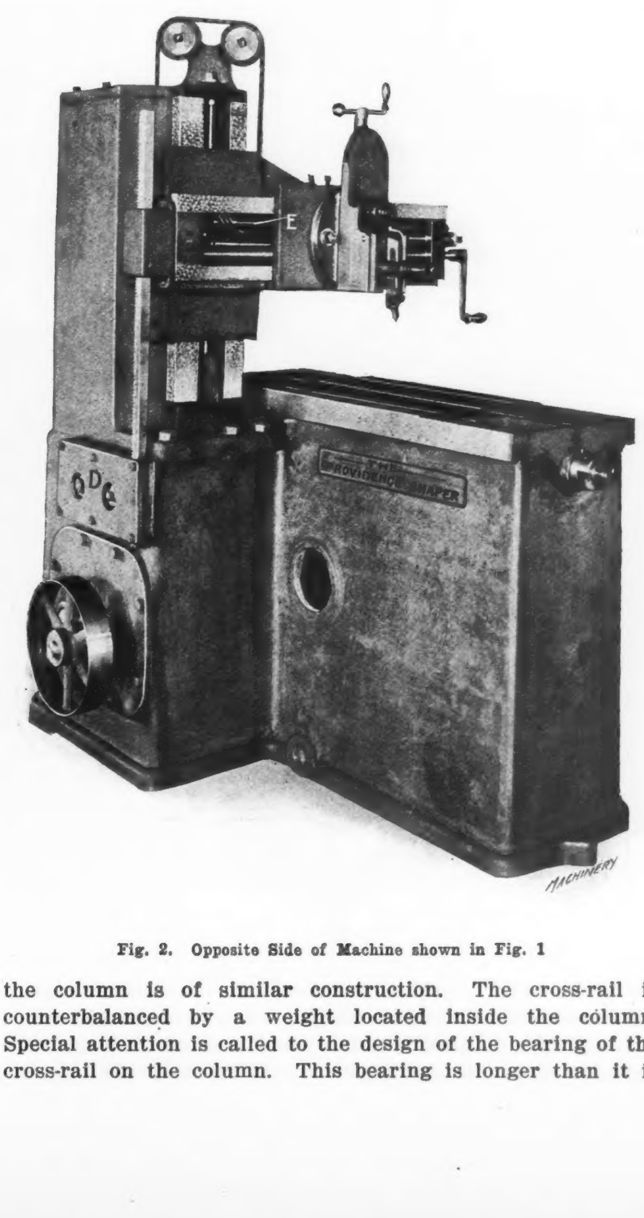


Fig. 2. Opposite Side of Machine shown in Fig. 1

wide in the ratio of $2\frac{1}{4}$ to 1. The cross-rail is secured in any desired position by drawing in the spring gib by means of a clamping bolt located at the back of the cross-rail. Tightening this bolt draws the cross-rail back against the face of the column and eliminates the chance of any play between these two members.

The positioning of the table is secured by means of a crank operated adjusting shaft *A* at the side of the table. This shaft works in conjunction with a pair of spiral gears which turn the main adjusting screw *B* beneath the table. This screw turns in a nut that moves the table to the desired position. The nut is gibbed in placed so that it acts as a support for the screw, the construction being such that there is no chance of deflection.

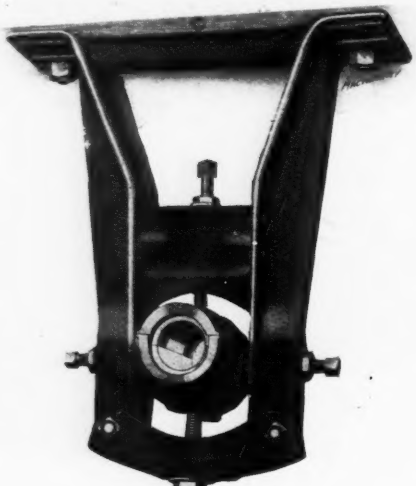
The motion is transmitted to the feed shaft through the rocker arm *C*, Fig. 1, the rate of feed being regulated by the star wheel. The motion is transmitted by a pair of spur gears located in the case *D* at the opposite side of the machine to the rack operating gear. By this means, the rack is operated to transmit the motion to the feed shaft running through the cross-rail. The feed operating gear is located on the opposite end of this shaft, and this gear transmits the motion to either the cross or vertical feed-screws by a sliding gear. This arrangement will be better understood by referring to the line cut, Fig. 3. The cross-rail is raised by a crank operated shaft *E*, Fig. 2, the position of which is fixed at the top of the cross-rail. This shaft provides for quickly raising or lowering the cross-rail to any desired position, the motion being transmitted to the vertical elevating screw through a pair of spiral gears.

The gear-box is of the unit type construction. By loosening the bolts on the flange of the case, the entire gear-box may be removed with the gears intact. The inner end of the gear-box is fitted over the end of the driving shaft bearing bushing, giving adequate support and insuring accurate alignment. The gears are of wide face so that ample strength is provided.

The principal dimensions of the machine are as follows: Stroke of table, 24 inches; horizontal travel of tool, 19 inches; vertical feed of tool, 6 inches; vertical adjustment of cross-rail, up to 16 inches; travel of saddle on cross-rail, 19 inches; distance from table to cross-rail when up, 16 inches; number of changes of table speed, 6; strokes of table per minute, 10 to 100; ratio of power gearing (maximum) 60 to 1; ratio of power gearing (minimum) 6 to 1; net weight of machine, 4400 pounds. Mr. James Coulter, vice-president of the Providence Engineering Works, Providence, R. I., is the designer of this machine.

STANDARD STEEL SHAFT HANGERS

The accompanying illustration shows the 1914 model of the "Pioneer" steel shaft hanger manufactured by the Standard



The "Pioneer" Steel Shaft Hanger

Pressed Steel Co., Philadelphia, Pa. The construction of this hanger includes features which represent the result of an experience of ten years in the manufacture of steel hangers and, in addition, several new features have been added. The use of steel in hanger construction insures a high degree of strength, which not only adds to the durability of the hanger but is also a factor in eliminating industrial accidents. Another noteworthy feature is the ease with which this type of hanger may be erected owing to the fact that the frames weigh much less than the corresponding size of hangers made of cast iron.

LENNOX SERPENTINE SHEAR

The accompanying illustration shows the Lennox serpentine shear that is a recent product of the Lennox Machine Co., for which Joseph T. Ryerson & Son, 16th and Rockwell

Lennox Shear having a Capacity for cutting Material up to No. 10 Gage

Sts., Chicago, Ill., have the sales agency. This machine is particularly designed for both straight and irregular cutting operations on sheets and plates. The frame is a steel casting of spiral construction designed to provide the necessary clearance when cutting material of unlimited length or width. Both straight cuts and "in and out" curves with a minimum radius only slightly larger than the diameter of the shear blades can be handled. All of the gearing is carried by the spiral steel frame which is mounted on a substantial base. All gears have cut teeth, and cast-iron gear guards are provided to avoid the possibility of accidents.

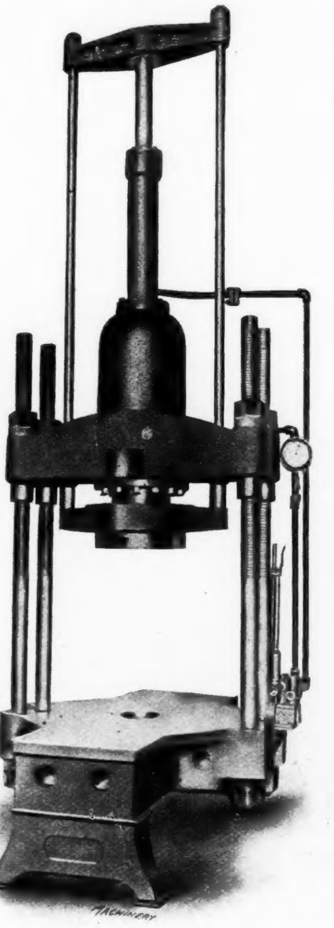
The blades are made of high-grade tool steel and set in such a way that there is very little distortion of the work in cutting. The upper cutter is positively driven while the lower cutter is mounted on an adjustable sleeve so that its position may be varied in cutting material of different thicknesses. There is also a cam movement provided which enables the lower cutter to be dropped sufficiently to permit the removal of sheets without reversing the machine. The cutters have a flush fastening on their shafts with no nut projecting out to interfere with the work. The knurled edges of the cutters feed the sheet automatically into the machine. A tool steel pin is provided to take the end thrust on the lower cutter shaft.

Where a number of sheets are to be cut to the same pattern, a templet may be bolted to the work; such a templet is then followed by guiding it against the top cutter. It will be seen that the machine is driven by means of a two-speed

pulley which gives a slow speed for intricate cutting and a high speed for simpler work. The main driving shaft is extended and squared on the end so that a crank may be used for turning the machine by hand if power is not available. The shear shown in the illustration has a capacity for cutting material up to No. 10 gage. Other sizes of these machines are made with capacities for cutting material up to 16 gage, up to ¼ inch and up to ⅜ inch. All machines are arranged for either belt or hand power, or for direct-connected motor drive.

HYDRAULIC STRAIGHTENING PRESS

The accompanying illustration shows a 250-ton hydraulic straightening press recently built by the Hydraulic Press Mfg. Co., 84 Lincoln Ave., Mt. Gilead, Ohio. This machine is primarily intended for straightening structural steel shafts and bars, but it can also be employed for a wide range of work in the machine shop, being particularly suitable for bending, forcing, broaching and similar operations. The stock space can be varied from a minimum of 20 inches to a maximum of 48 inches. It will be seen from the illustration that the strain rods are threaded on their upper end and provided with nuts on both the upper and lower sides of the cylinder bearings.



Hydraulic Straightening Press

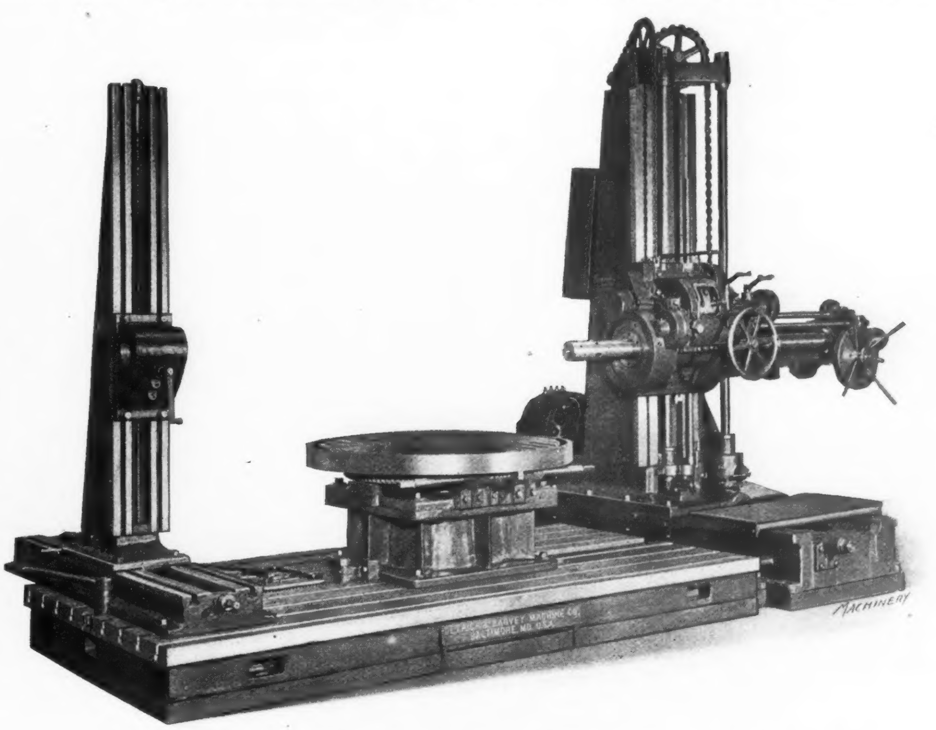
When it is found desirable to increase the stock spaces of the press a block of some suitable material is placed between the ram and the pressure bed. The upper nuts on the strain rods are then set to the desired point and the pressure applied. This action raises the cylinder until its bearings come in contact with the nuts. When the work is of such a character that it requires a smaller space, the block is still kept between the ram and pressure bed. In this case, the lower nuts are adjusted to the desired point while the pressure is still on. The pressure is then released, thus allowing the cylinder to drop to the desired point. The upper nuts are then screwed down against the cylinder bearings. In order to provide for raising and lowering the cylinder, the pipe connections have swing joints. T-slots are provided in both the ram head and bed of the press for use in attaching bending blocks, forms or dies. The body of the press provides for holding the work for bending between centers having a maximum distance of 72 inches.

It will be seen that the bed has a hole in the center which permits the press to be used for broaching, for forcing wheels on or off shafts and for similar operations. The diameter of the ram head is 20 inches and it has a run of 20 inches. The length of the press bed over all is 72 inches, while the width of working space is 26 inches. Steel is used throughout its construction. The pressure capacity of this press is 250 tons.

The four-way poppet operating valve applies the pressure to either the main pressure ram or the return ram in the pull-back cylinder, and at the same time permits the water or oil used in operating the press to return to the pump reservoir from the cylinder on which the pressure is released. This valve is operated by a single lever which has three positions. In the first position, the pressure on the main pressure ram is applied and the pressure released on the return ram. In the second position, the pressure is applied on the return ram and released on the main ram. In the third or neutral position, the pressure is held with the rams at any point in their travel in a stationary position. A five-way, high and low pressure, double-acting valve can be furnished if desired.

DETRICK & HARVEY BORING MACHINE

The illustration shows a standard No. 2 boring machine built by the Detrick & Harvey Machine Co., Baltimore, Md., which is equipped with a thread chasing attachment and a rotary table with a removable sub-base. The standard form of outboard bearing is also provided. The removable sub-base is shown in position beneath the rotary table, but when so desired this sub-base may be removed and the rotary table placed directly upon the floor plate of the machine.



Detrick & Harvey Boring Machine with Thread-chasing Attachment and Rotary Table

The thread chasing attachment is driven from the feed and is located on the rear of the saddle horn. The design is such that the chasing attachment cannot be thrown into action at the same time that the spindle is feeding. The arrangement of the change gears is similar to that used on a lathe and can be furnished to cut any required thread.

The rotary table is 60 inches in diameter, the rotary motion being hand operated. As previously stated, the sub-base can be removed from beneath the table to enable the table to be brought down closer to the floor plate. The height from the top of the table to the floor plate is 27 inches with the sub-base in position and 15½ inches when the table is set directly on the floor plate. The table can be moved along the floor

plate by means of a rack and pinion operated by a lever. The rotary table is graduated to ½ degree to facilitate accurate setting.

BROWN CONTINUOUS RECORDING INSTRUMENT

The Brown Instrument Co. and the Keystone Electrical Instrument Co., Philadelphia, Pa., are placing a continuous recording instrument on the market for use as an electrical pyrometer. The instrument can also be employed as a recording volt-meter or ammeter. In its design, particular care has been taken to produce an instrument of as simple construction as possible, and at the same time to make it of the most compact form. The case of the instrument is only 15 inches high by 8 inches wide, and it projects out from the wall or switchboard a distance of 7 inches. The space occupied has been considerably reduced by placing the clock mechanism behind the record chart instead of to one side of it; the record chart is the only part shown on the face of the instrument.

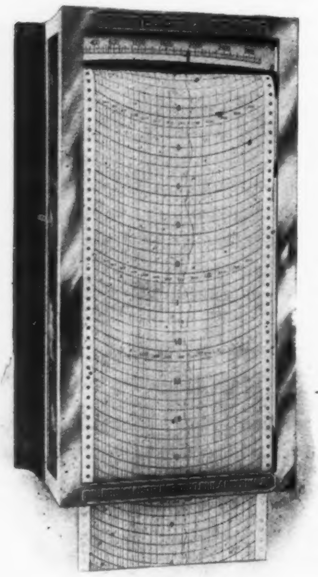


Fig. 1. The Brown Continuous Recording Instrument

A two month roll of record paper is used, which can be supplied with an inked ribbon to make a dotted ink record, or the record can be made on coated paper which entirely does away with the use of ink. It is only necessary to wind the clock once a week and to change the roll of record paper once in two months. The instrument in Fig. 1 shows a ten-hour record on the chart and a scale is placed above the record so that the indications are easily read at all times. The paper also has the scale printed on it for direct reading. Fig. 2 shows an instrument with the door open so that the interior construction may be more clearly seen. It is only necessary to open the case once every two months when the roll of record paper is changed.

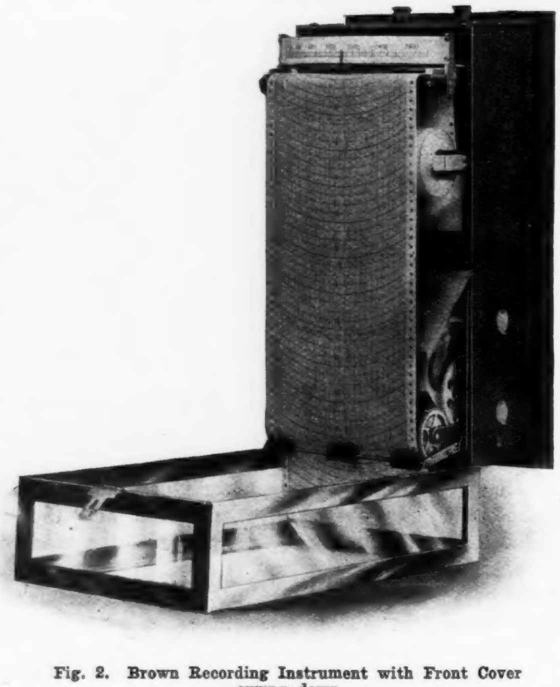


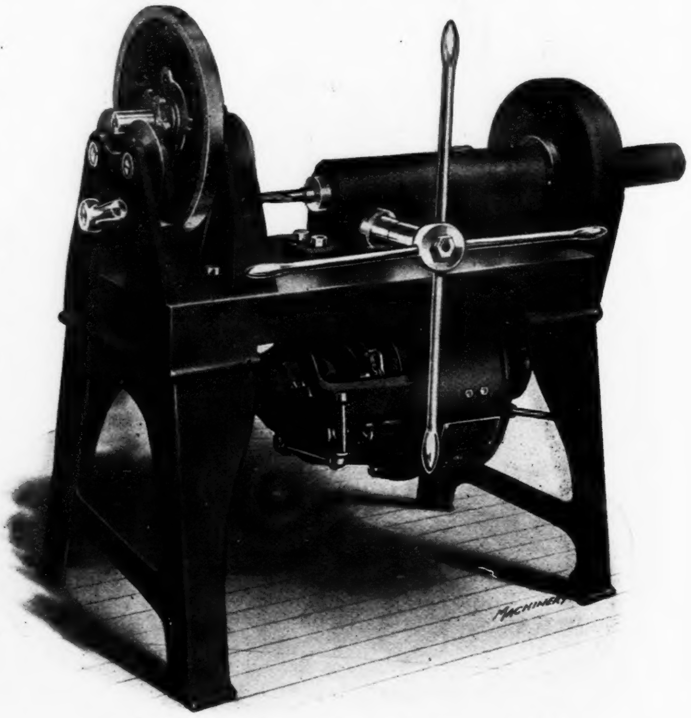
Fig. 2. Brown Recording Instrument with Front Cover swung down

This recording instrument will prove particularly useful as an electrical pyrometer used in conjunction with thermocouples, for resistance thermometers for measuring temperatures, or for recording the voltage or current in an electrical circuit. As the d'Arsonval type of direct-current instrument used is of the frictionless type to prevent lag in the readings, it will be seen that a particularly sensitive and accurate instrument is secured.

ROCKFORD COMBINATION DRILLING AND BALANCING MACHINE

Where the practice of balancing flywheels, armatures, and other rotating parts by removing the stock by drilling is employed, a great amount of time is lost in transferring the work from the drill to the balancing ways in order to determine whether the required balance has been secured. The accompanying illustration shows a combination balancing and drilling machine which has been developed by the Rockford Tool Co., Harrison Ave. and 11th St., Rockford, Ill. This machine eliminates the necessity of moving the work back and forth between the drill press and balancing ways, after each hole has been drilled, and as cases are not uncommon where eight or ten holes have to be drilled before the required balance is secured, the saving effected will be readily appreciated.

The arrangement of the machine will be easily understood from the illustration, where it will be seen that the part to be balanced is carried on an arbor supported on rotating disks. These disks are hardened and ground and the pivots run on ball bearings which insures a very sensitive movement. The work can be clamped in place and a hole drilled, after which it is released and the balance tested. Subsequent drilling operations are performed by merely retightening the work, this operation being repeated until the required balance is secured.



Rockford Combination Drilling and Balancing Machine

SIMONDS IMPROVED HACKSAW BLADE

A new "hard edge" flexible hacksaw blade has been put on the market by the Simonds Mfg. Co., Fitchburg, Mass. The chief feature of this blade of importance to users is that it will do the cutting work of the regular hard blade, and in addition is so flexible that it can be bent a great many times without breaking, thus eliminating a large percentage of the breakages that every user of hacksaw blades expects. The Simonds "hard edge" flexible blade is made in the standard lengths and numbers of teeth per inch.

POTTER & JOHNSTON AUTOMATIC PISTON AND PISTON-RING MACHINE

The Potter & Johnston Machine Co. of Pawtucket, R. I., has recently developed a special automatic piston and piston-ring turning machine, known as the No. 2 P. This machine is intended for use by automobile manufacturers in turning eccentric and concentric piston-rings, as well as in turning the pistons themselves. Any diameter of piston from three to eight inches and up to twelve and a half inches long may be turned, and piston-rings of any width and from three to eight inches diameter are turned and cut off automatically from the piston-ring sleeve. The machine is entirely automatic in operation, the attendant simply placing the piston on the draw-back arbor and starting the machine. The turning and grooving are then automatically done. Similarly, on piston-rings, the piston-ring sleeve casting is held in the chuck and the entire casting turned up, boring out the center and turning the outside eccentric, at the same time marking each ring at the thin spot for separating, as well as cutting off the entire number of pistons automatically. A general view of the machine is shown in Fig. 1, while in Fig. 2 the machine may be seen as viewed from the rear. In both these views the machine is shown set up for piston-ring turning. Figs. 3 and 4 are plan views of the machine with the tooling equipment necessary for piston-ring turning and piston turning, respectively.

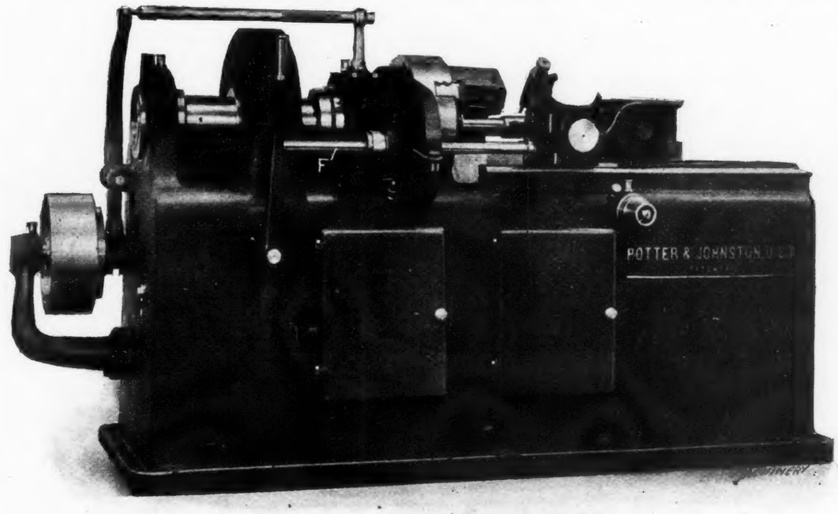


Fig. 1. Potter & Johnston Automatic Piston and Piston-ring Machine

From Fig. 1 it will be seen that, in general, the machine resembles the Potter & Johnston automatic chucking machine, except that it is simpler, the different feeds, speeds and attachments being eliminated from this machine, and everything is done to make it extremely powerful and simple on the operations for which it is intended. Power is carried to the driving shaft by a thirteen-inch single pulley drive. The driving shaft is supported by an outboard bracket and also fitted with a friction clutch so that the machine may be started and stopped independently of the drive.

As may be seen in Fig. 2, the gear-box from which the machine is controlled is at the rear and by its means the forward feeding of the tools is effected; a quick return movement to the tools in the ratio of 65 to 1 is provided for, and also the gears may be put into a neutral position so that no feeding of the cutting tools will take place. The general design of the gear-box is the same as that employed on all Potter & Johnston machines, but is much simpler because there is not the range of feeds and speeds to be provided for. The forward feeding of the tools is accomplished by planetary gearing in which there are two gears side by side varying by one tooth, and meshing with them is a planetary pinion that travels around these two gears, gaining a tooth at each revolution, thus causing one of the gears and hence the feed to advance by this amount. When in the neutral position, no feeding taking place, the ratchet and latch that gives the feed are disconnected, and the position is called the "latched-up" position. For the return motion, a clutch is thrown in, connecting the back-shaft directly with the reverse gear and causing the tool to return at a speed sixty-five times greater than the advance.

From the intermediate shaft, gearing transmits power to the main spindle directly through a large driving gear. This gear is encased as may be seen in the photograph. The driving spindle is 5¾ inches diameter and has long bearings, giving it ample power and long life. The spindle speed is constant and fixed according to the diameter to be turned, any variations in speed being obtainable through change gears. The oiling system is located over the gear-box (see Fig. 2) and from the central reservoir, tubes run to the important bearings. Each tube is fitted with an independent regulator to govern the supply of oil to the individual bearings.

The back-shaft, which may be seen at *A* in Fig. 2, transmits motion to a cross-shaft through bevel gearing, and then by a worm and worm-wheel to the turret drum-shaft. A friction clutch is located beyond bearing *B* so that in case of excessive strain on the cutting tools, the friction clutch will slip and save the mechanism from injury.

The turret drum-shaft and turret drum are, of course, within the machine and directly under the turning slide *C*. The cam-plates on this turret drum are of hardened steel and thus have a long life. The turning slide corresponds to the ordinary turret slide, but does not rotate as in the chucking type of machine, its working face always being presented to the work. The turning slide carries the cutting arbor *D*, and actuates its forward movement. The rocker arm *E* upon which the cutting-off tools are mounted is actuated by a face-cam on the turret drum-shaft.

At the front of the machine is the eccentric driving shaft *F*. This is driven by a pair of spur gears from the main driving spindle, and transmits motion to the eccentric turning slide that provides the motion for the eccentric turning. Referring now to the plan view, Fig. 3, the operation of the tools may best be observed. Here the eccentric driving shaft is shown at *F*, working in an eccentric bushing *G* that gives the in and out motion through a vertically operating slide *H* to the cutting tool slide *I*. On the end of this cutting tool slide is the eccentric turning tool *J*. Mounted at the left of the eccentric tool slide is the automatic marker that locates the thin point on each ring and marks it so that it can be correctly cut. This marker bracket is shown at *K*, and the marking plunger *L* is operated through a connection with lever *M* which is, in turn, operated from cam *N*. This cam follows the eccentricity of the piston-ring being turned and has an abrupt drop at the thin point, allowing the marker plunger *L* to come in quickly and make an impression upon the piston-ring at the low point. A spring in this marking tool is, of course,

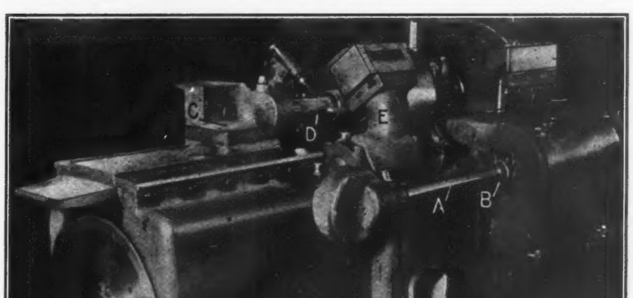


Fig. 2. Opposite Side of Machine shown in Fig. 1

necessary to cause the tool to cut into the ring. In addition to being held and supported by the turret slide, the eccentric turning tool and its bracket and the automatic marker are well supported by a bearing plate *O* which has a continuous bearing upon the steel plate that is bolted to the top of the front way of the machine.

The operation of the machine when working on piston-rings is as follows: The piston-ring sleeve casting is held in machine and the thirteen rings are turned, marked and cut off in 4½ minutes.

The cutting-off tools carried by the rocker arm are first adjusted and set in position in a removable block that is, in turn, held on the rocker arm. This block, with the tools intact, can easily be removed at any time for resharpening and the block returned without affecting the adjustment of the machine in any way.

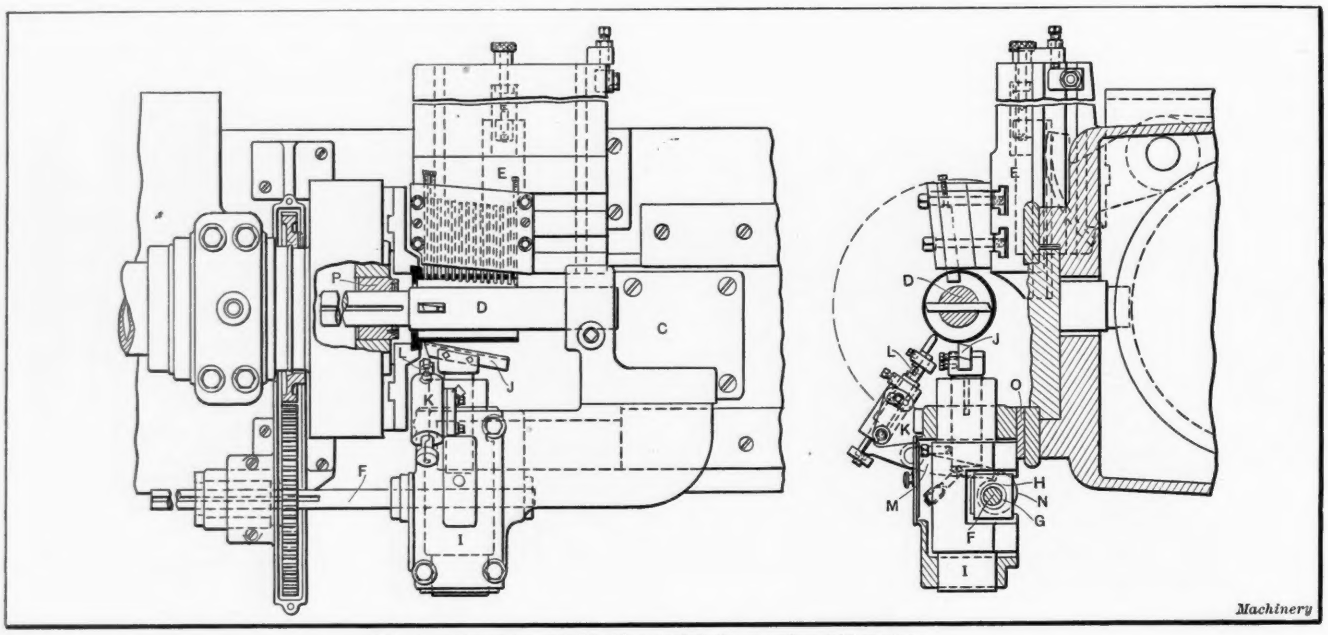


Fig. 3. Plan View of Machine tooled for turning Piston-rings

the universal chuck upon the main spindle of the machine. The boring bar is held on the turret slide and its inner end is piloted in the bushing *P* which may be seen in the line illustration, Fig. 3. This bushing is fitted with a felt washer which keeps dust and chips out of the bushing. The concentric boring tool is, of course, located in the boring bar as shown. The eccentric turning tool is firmly bolted in the eccentric turning slide and receives its motion in the manner just described. As soon as the eccentric turning tool has

When working on pistons, the tool set-up is about as shown in Fig. 4. Here the piston is shown at *A*, being held in place upon the spindle nose *B* by means of a cross pin *C*, passing through the wrist-pin hole, which is drawn back by a draw-back bar *D*. The draw-back bar is quickly operated by means of lever *E*. The pistons are bored and turned on the open ends before coming to this machine. When used for piston-turning, no eccentric turning slide is necessary, so the bracket supporting the eccentric turning and marking fix-

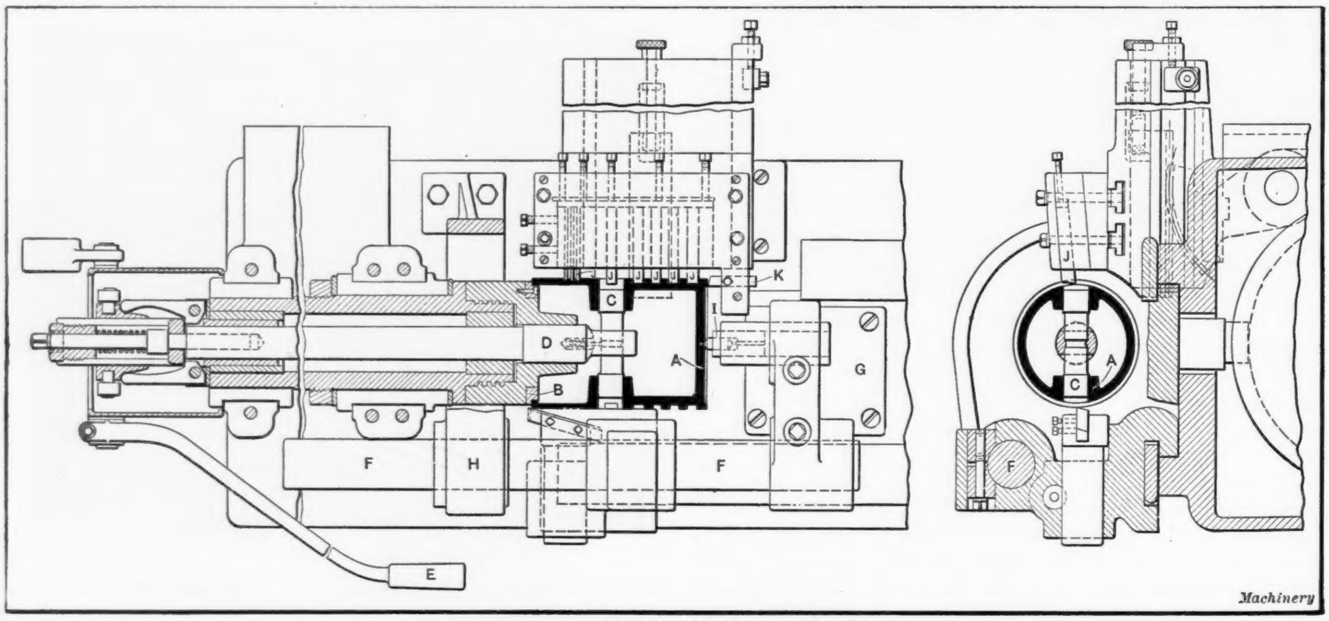


Fig. 4. Plan View of Machine tooled for turning Pistons

commenced its cut, the rocker arm with its multiple set of cutting-off tools comes in from the rear. This is clearly shown in Fig. 3, and as the cutting tools are set each slightly ahead of the other, the rings are separated about as soon as the outside portions have been turned and marked. An idea of the speed at which piston-rings are made may be gathered from the fact that a casting which will make thirteen piston-rings, 5/16 inch wide and 4 1/4 inch diameter, is put into the

tures is removed. The turning bar *F* is operated from the turning slide *G* and its outer end is piloted in a bracket *H* bolted to the frame of the machine. By thus piloting the turning tool, great stiffness is secured. A centering tool is located at the front of the turning slide. The grooving tools *J* and the end turning tool *K* are operated from the rocker arm, these being substituted for the cutting-off tools used in piston-ring turning. As in the case of piston-ring turning, the

different operations on the pistons are performed simultaneously, being centered, faced, turned and grooved at the same time.

In accordance with recognized practice, pistons are finished on this machine in two operations, roughing and finishing, sufficient time being allowed to elapse between these operations to insure accuracy. The turning time for pistons, 3¾ diameter by 3¾ inch long, is 6 minutes. This includes roughing and finishing.

STANDARD ROLLING MILLS

The accompanying illustrations show two rolling mills which constitute an addition to the line of machines manufactured by the Standard Machinery Co., Elmwood Ave., Auburn, R. I. The machine shown in Fig. 1 is a light mill with rolls 5 inches diameter by 8 inches face width. This machine is equipped with a constant-speed motor used in connection with a special Cutler-Hammer controller which allows the mill to be started and stopped by a foot treadle.

Beneath the bed of the machine there is an oil switch which is connected to an automatic switch shown in the illustration; this arrangement allows the operator to press down and lock the treadle on the special treadle stand which has a shoulder on it. When the rolling operation is finished,

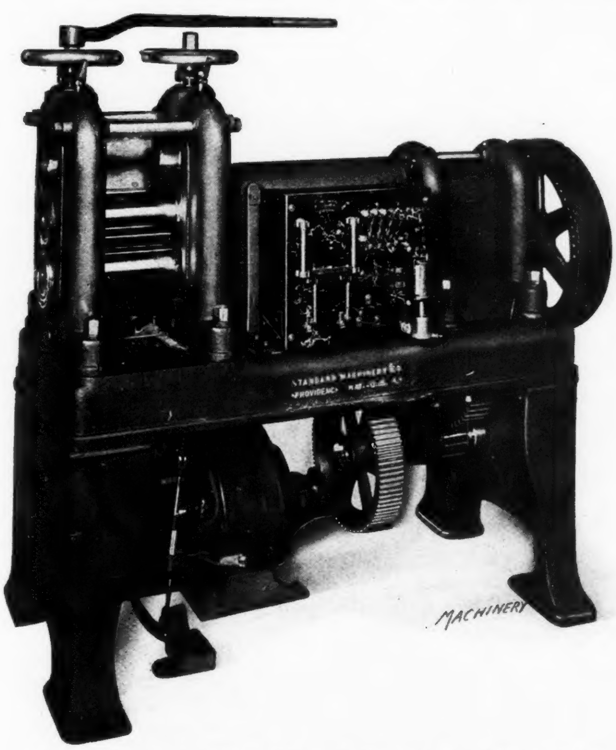


Fig. 1. Standard Rolling Mill with Constant Speed Motor Drive

the treadle is pushed to one side, allowing it to snap back through the action of a spring and thus shut off the current from the motor. Exclusive of the motor train, the machine is driven through a triple train of gearing which includes the intermediate and driving trains on the machine proper with cut steel herringbone gears in the housings. The motor pinion is of rawhide and meshes with a cut steel gear on the driving shaft. The teeth of all of the gears are cut.

The machine is equipped with hardened and ground rolls of special Krupp steel. These rolls have hardened and ground journals which run in the Standard roller bearings. The face of the rolls are mirror lapped. The housings are fitted with handwheels with micrometer dials. The principal dimensions of the machine are as follows: Sizes of rolls, 5 inches diameter by 8 inches face width; back gear ratio, 16 to 1; floor space occupied, 7 by 2½ feet; weight of machine, 3600 pounds.

Fig. 2 shows a machine which is a modification of the rolling mill illustrated in Fig. 1, the former machine being arranged for belt drive. The special features of this machine consist of making the bed all one piece and doing away with the legs and bearing bracket below the separate bed. This construction makes a self-contained machine of the same dimensions as the one previously described. All of the bearings are bronze bushed. The clutch is of the cone type and permits the mill to be started and stopped without requiring the operator to use his hands, the clutch being

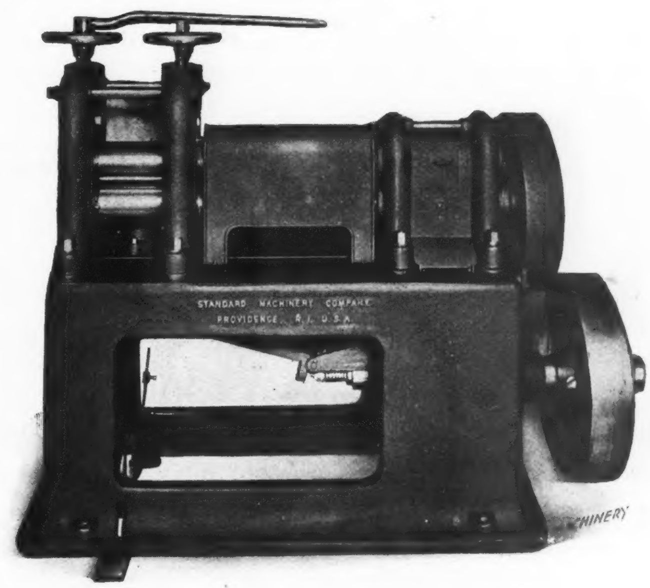
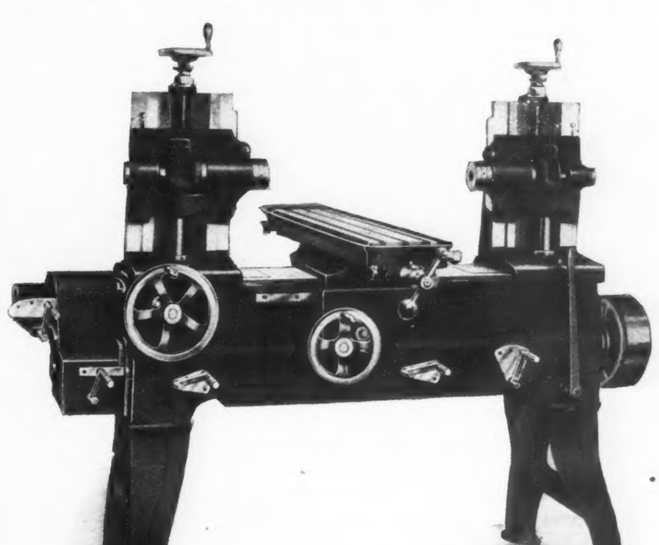


Fig. 2. Same Type of Mill arranged for Belt Drive

thrown in and out by means of a spring mechanism on the driving shaft.

KNIGHT DUPLEX MILLING MACHINE

The light duplex milling machine illustrated herewith is a recent production of the W. B. Knight Machinery Co., 2019-25 Lucas Ave., St. Louis, Mo.. This machine has been designed for manufacturing work where it finds application on



Knight Duplex Milling Machine for Light Work

many light jobs that can be milled on two sides simultaneously. The duplex machine cuts the operating time exactly in half, and as a second setting of the work is saved, the chucking time is also reduced 50 per cent. Among the classes of work which this machine will handle, the following may be mentioned: milling the frames of typewriters, adding machines, cash registers and similar parts where lugs occur in pairs and are to be finished on opposite sides of the casting; and milling flats, squares and hexagons on shafts, tools, etc. The work may either be handled by putting a number of

pieces vertically in a jig and passing them between two end mills to finish opposite sides simultaneously, or by bringing the heads closer together and driving two cutters in opposite directions with one cutter above the other.

As the machine is particularly designed for the class of work which has been referred to, the longitudinal feed of the table is the only one required. The table is provided with a quick return at 7½ times the feed. The feeds and speeds are obtained through gears which are fully enclosed, and the feed and adjustment screws are fitted with graduated collars. The spindles run in adjustable bronze bearings. Both the uprights carrying the spindles are controlled by one handwheel which moves them toward or away from each other simultaneously. This handwheel has a graduated set collar to indicate the distance that each head has moved. The heads can be raised or lowered on the uprights, independently of each other and graduated scales on the vertical ways and graduated set collars on the vertical screws facilitate an exact adjustment which is matained by means of lock nuts. Both the uprights and saddle carrying the table are also clamped to the bed of the machine by means of lock nuts.

Gears contained in the bed of the machine provide six changes of spindle speeds, the same gear box controlling both of the spindles. The drive is through gears which transmit power from the single clutch pulley at the end of the bed. At the opposite end of the machine from the driving pulley there is another gear box which controls the longitudinal feed of the table. There are six all-geared feeds instantly obtainable, the feed range being from 0.008 to 0.060 inch per revolution. Both sets of change gears run in oil and all levers controlling changes of speeds and feeds are conveniently located at the front of the machine. Adjustable gibs are used to compensate for wear between the spindle heads and the uprights. The principal dimensions of the machine are as follows: maximum distance between spindles, 22 inches; height of spindles from bed, 2 inches to 9 inches; range of spindle speeds, 50 to 400 R. P. M.; working surface of table, 8½ by 32 inches; longitudinal table feed, 24 inches; floor space occupied, 73 by 73 inches; net weight of machine, 1550 pounds.

This machine is to be exhibited at the Boston Auto Show, Mechanics' Bldg., March 9 to 14.

WHITCOMB-BLAISDELL GEARED HEAD LATHE

The Whitcomb-Blaisdell Machine Tool Co., Worcester, Mass., has recently added to its line of lathes a geared head machine which is illustrated herewith. The changes of spindle speeds on this machine are obtained through sliding clutches which are so arranged that it is impossible to engage two speeds at the same time. The idea in bringing out these machines was to meet the demand for a powerful geared head lathe which is adaptable for either single pulley drive from the lineshaft or direct-connected motor drive through a constant-speed motor. The design combines the features of simplicity and durability.

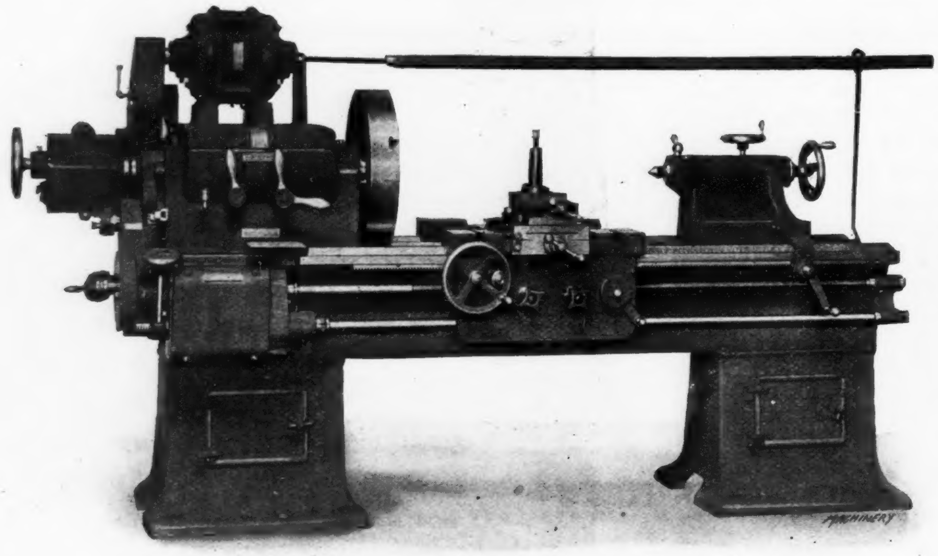


Fig. 1. Whitcomb-Blaisdell Geared Head Lathe with Single Pulley Drive

A noteworthy feature of these machines is that the spindle may be stopped, started or reversed from the operating position without requiring the use of a countershaft or reversing motor equipment. This feature is obtained through the application of two clutches and gearing located in the reversing friction gear box at the head end of the lathe. Although a single speed motor is entirely satisfactory it is, of course, possible to use a 2 to 1 or 3 to 1 variable speed motor, but the variable speed motor is by no means necessary. In addition to the reversing geared headstock, these machines include all of the regular features applied on Whitcomb-Blaisdell lathes. The line includes machines ranging from 14 to 30 inches.

ALLEN SOCKET FILLISTER SCREW

In the November, 1910, number of MACHINERY, the safety set-screw made by the Allen Mfg. Co., Inc., 135 Sheldon St., Hartford, Conn., was illustrated and described. This company has recently brought out a fillister-head screw which is turned by a key fitting in a socket in the head; one of these screws is illustrated herewith, and little description will be necessary to point out its features. This form of screw will stand up under hard service where the slotted head screws soon become so badly worn that it is very difficult to turn them. Anyone who has had any experience in turning ordinary fillister-head screws with a screwdriver knows that the slot often becomes battered in a way which makes the screw very difficult to turn.

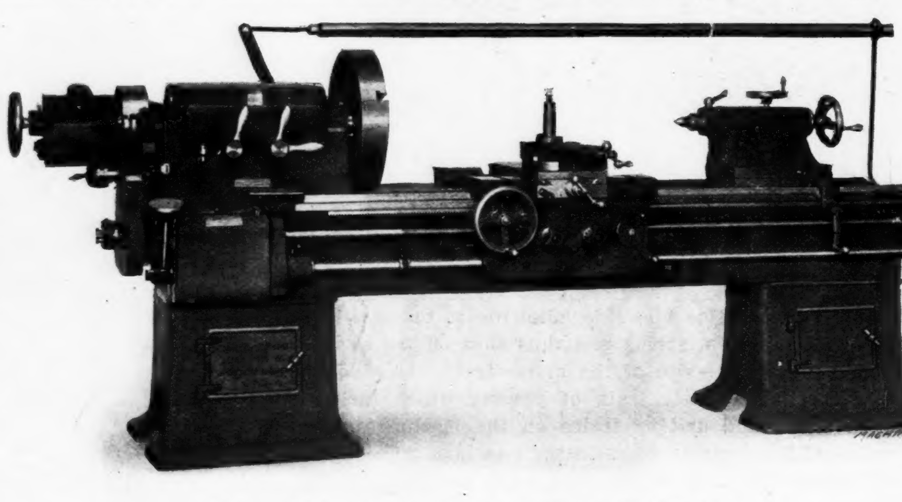
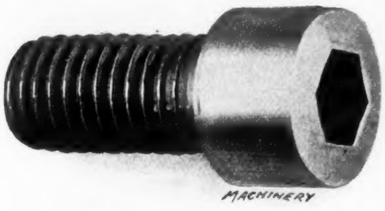


Fig. 2. Whitcomb-Blaisdell Lathe with Electric Motor Drive

The application of the socket head idea on fillister screws enables the screw to be turned up fully as tight as an ordinary

hexagon headed capscrew. As no clearance is necessary around the head, these screws can be conveniently used in many places where there would not be sufficient clearance for a hexagon headed screw. The heads of the Allen fillister screws are trimmed so that the head and body are in perfect alignment. Screws of this kind improve the appearance of a machine and constitute a good "talking point." At the present time the Allen Mfg. Co. is making screws of this kind in sizes ranging from 3/16 to 3/4 inch, and in any length. Other sizes will be added to the line from time to time.

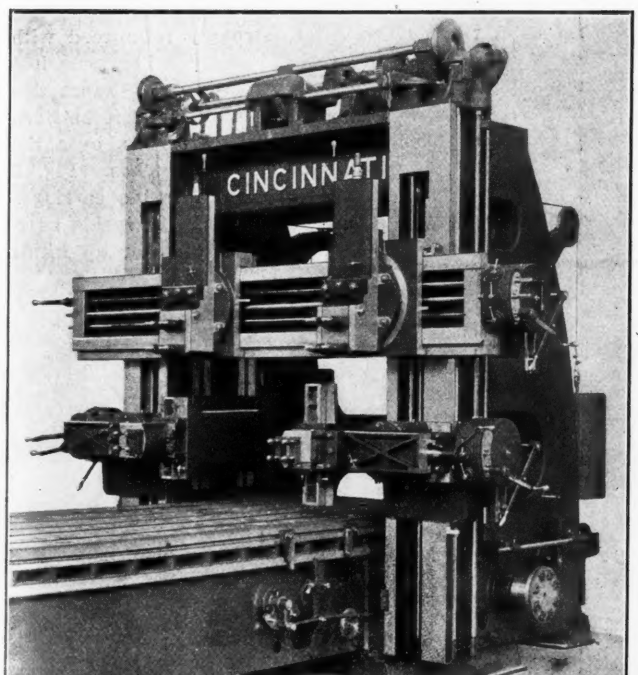


Allen Socket Fillister-head Screw

CINCINNATI LOCOMOTIVE CYLINDER PLANER

In the January, 1914, number of MACHINERY a large size planer built by the Cincinnati Planer Co., Cincinnati, Ohio, was illustrated and described. The accompanying illustration shows a machine of similar design which has been built for planing locomotive cylinders. This machine has a capacity of 72 by 72 inches and has rapid power traverse to all heads in any direction. All movements are independent of each other and can be operated whether the table is in motion or not.

The special feature of this machine lies in the provision of the auxiliary side head supports. The heads are provided



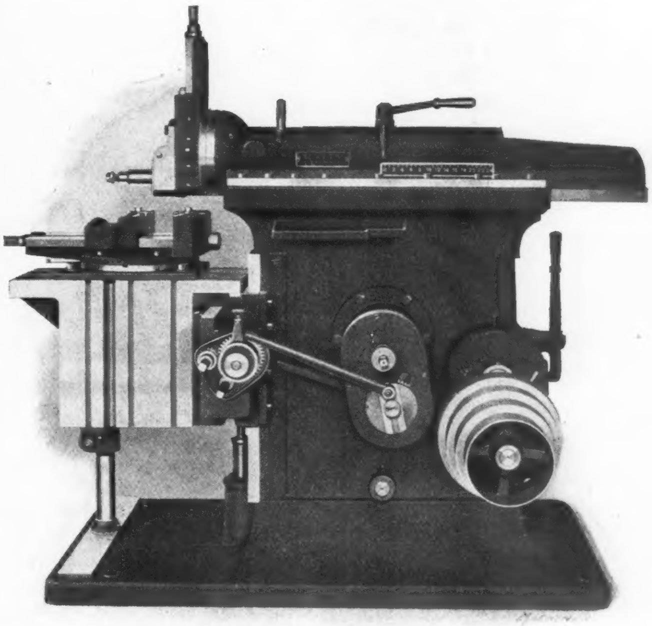
Cincinnati 72 by 72-inch Locomotive Cylinder Planer

with extra brackets which are rigidly bolted to the inside of the housing; these brackets support the slides while the machine is engaged in planing the frame fits and similar operations. They are adjustable for height and can be used in any position between the top of the table and the bottom of the cross-rail. A sliding shoe fits the front face of the brackets and has a dovetail on it which the cross-slide fits. This construction eliminates all twisting strains on the face of the housing caused by the long overhung slide and the upward pressure of the tool. These special brackets can be easily removed from the machine, thereby converting it into a standard planer. This planer is built in various sizes and lengths.

R. A. KELLY SHAPER

The accompanying illustration shows a shaper which constitutes a recent addition to the line of machines manufactured by the R. A. Kelly Co., Xenia, Ohio. The feed on this machine is through a ratchet and plunger mechanism, the ratchet being covered over the greater part of its circumference. The stroke of the pawl is always the same amount, so the feed depends upon the number of ratchet teeth that are exposed. The ratchet cover is prevented from turning, through the action of a spiral spring that holds it just tight enough so that it can be turned by hand.

The shaft which carries the ratchet cover is fastened to a



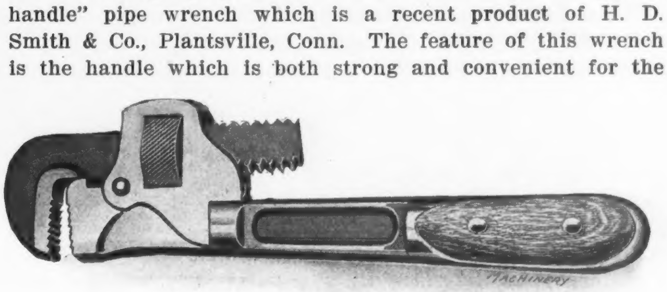
R. A. Kelly Shaper

radius rod, the other end of which is fastened to the radial gear guard carrying the lower feed gear. When the cross-rail is raised or lowered, the shaft carrying the ratchet cover rotates just enough to keep the same number of teeth in the ratchet exposed. In this way the cross-rail may be lowered from the highest to the lowest position or *vice versa* and no adjustment is needed.

The advantages of this feed mechanism are that it may be started, stopped, changed from a maximum to a minimum or reversed when the machine is in motion, without requiring any moving part to receive attention from the operator.

SMITH PIPE WRENCH

The accompanying illustration shows the Smith "perfect handle" pipe wrench which is a recent product of H. D. Smith & Co., Plantsville, Conn. The feature of this wrench is the handle which is both strong and convenient for the



Smith "Perfect Handle" Pipe Wrench

operator to hold. The swell at the end of the handle prevents the hand from slipping off and the wooden inserts make it comfortable to use in extremely hot or cold weather.

BRIDGEFORD AXLE LATHE

The accompanying illustrations show a center drive gap axle lathe with two carriages, which the Bridgeford Machine Tool Works, Rochester, N. Y., has recently designed for use

in railroad shops. This machine is intended for turning journals of car axles without removing the wheels, and as it is equipped with two carriages, both ends of the axle may be finished simultaneously. This means a considerable saving of time taken to machine a pair of journals. In other respects, the construction of the present machine is similar to that of the regular line of axle lathes built by this company.

The most satisfactory method is to place the machine in a pit so that the axles with the wheels attached to them can be placed on the centers with very little trouble. The driving gear, which is two-pitch with a face width of 5 inches, is completely encased and runs in heavy bronze bearings scraped to an accurate fit. It is furnished with drivers for driving the axle, these drivers operating on the same principle as the self-centering style drivers furnished with the standard Bridgeford axle lathes. The driving gear is in two pieces, tongued and grooved together and secured by four heavy hinged bolts and nuts. These bolts are easily operated by means of a socket wrench and less than one-half turn of each nut is sufficient to release the gear so that it may be opened to allow the entrance of the wheels and axle. The upper half of the driving head operates on a heavy hinge stud in the front of the machine. To lift the head, the pull pin is pushed into place and the nuts are released. An eye-bolt is placed on the upper half of the head and a rope with counterweights is used to lift and lower the head.

The machine is driven by a constant-speed pulley and there are three geared changes of cutting speed. The gears are of steel with cut teeth and run in oil, the changes of speed being accomplished by shifting levers conveniently located on the case. From the speed box, the power is transmitted to the driving head by a shaft located inside the frame of the machine. All driving shafts are of high-carbon steel and run in scraped bronze bearings.

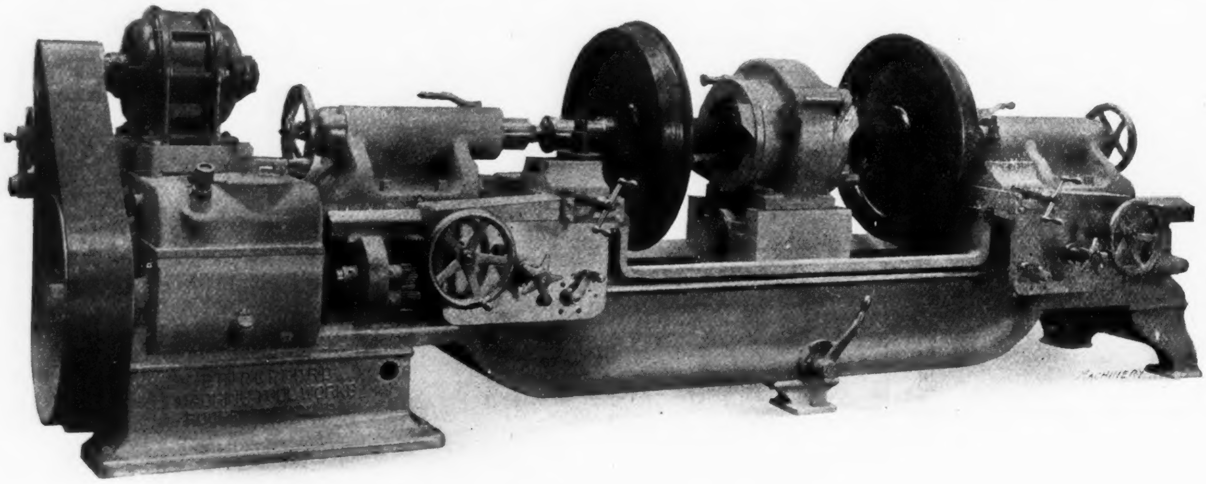


Fig. 1. Bridgeford Axle Lathe equipped with two Carriages

There are four instantaneous changes of feed ranging from 1/16 to 3/16 inch per revolution of the axle. The gears in the feed box are of steel and run in oil; they are operated by a lever placed at the center of the lathe. The tailstocks have bearings 24 inches in length on the bed; they are secured in position by four heavy bolts and binders and the spindles may be adjusted by means of screws and handwheels. The carriages are driven by a splined feed shaft and the direction of the feed is changed at the apron. Each carriage is independent of the other. The carriages have a bearing 30 inches in length on the vees and they also have a bearing at the back of the bed which takes the forward thrust, thus overcoming the tendency to raise the carriages from the vees when the burnisher is used.

The bed is of rigid construction, strongly reinforced with cross ties to provide the required rigidity. The principal dimensions of the machine are as follows: Minimum distance between centers, 54 inches; maximum distance between centers, 105 inches; swing over ways, 27 inches, over the carriages, 13½ inches and in the gap, 45 inches. Fig. 2 shows a machine equipped with two extra inside carriages for refinishing locomotive and tender axles with inside bearings. Such a machine is exceptionally useful for all kinds of axle repair work. The weight of a lathe equipped with two carriages is 16,000 pounds and a lathe with four carriages has a net weight of 18,000 pounds.

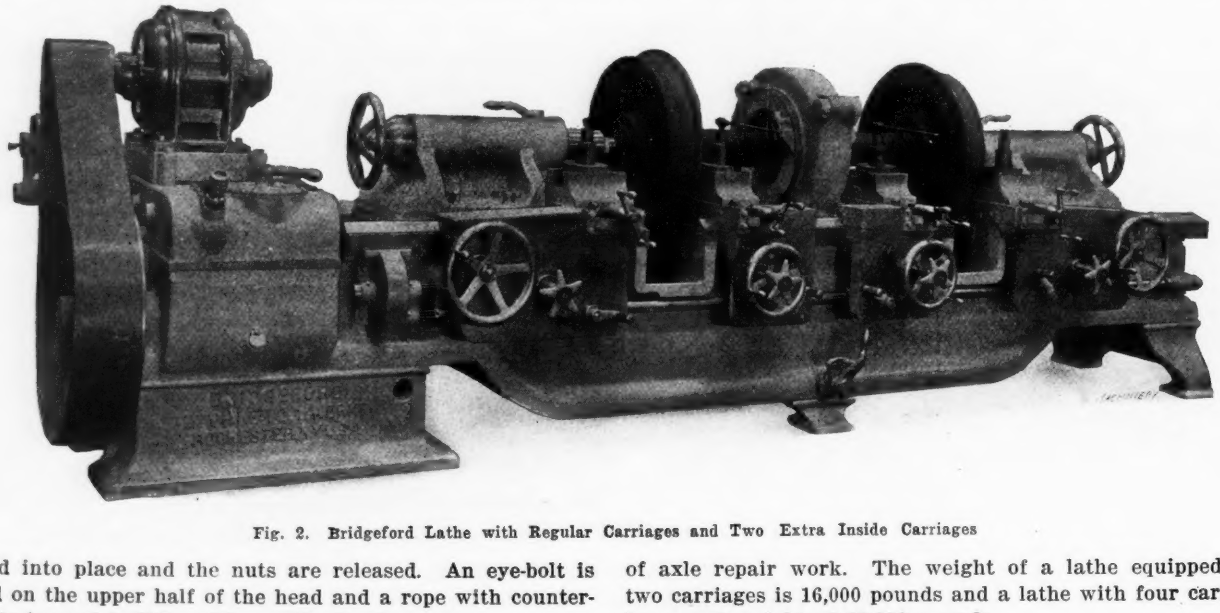


Fig. 2. Bridgeford Lathe with Regular Carriages and Two Extra Inside Carriages

PEDRICK PORTABLE MILLING MACHINE

The notable features of a portable milling machine which has recently been placed on the market by the Pedrick Tool & Machine Co., 3639 Lawrence St., Philadelphia, Pa., are the simplicity of its design, the adaptability of the machine for operation on a variety of classes of work, and the provis-

ion for driving by motor, belt or hand as desired. In a portable tool of this character, simplicity of design means less weight, greater convenience of operation and lower cost. For these reasons, automatic feed has been omitted on the No. 1 machine, hand adjustment for the longitudinal and cross traverse being provided, which serves every purpose. The No. 2 machine, which is considerably larger and intended for a wider scope of work, has automatic feed.

In designing this machine, particular care has been taken to have the construction of ample strength. The spindle bearing is proportioned to give rigid support and to have the

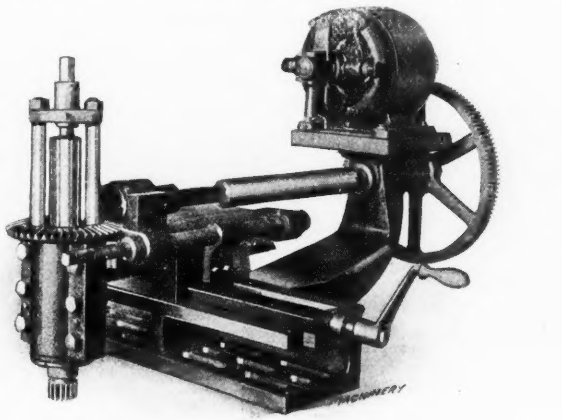


Fig. 1. Pedrick Portable Milling Machine

desired durability; the bed is ribbed in the center and has removable posts at both ends. These posts may be taken out if necessary when setting up the machine, as there are cases when they would interfere with a wrench used for tightening the holding nuts inside the base. There are a number of elongated slots in the base to provide for attaching the machine to the work. An important feature of the design is that the machine will mill a surface level with the surface on which it is attached or several inches below it. The machine does not "smother" the job and it will face a small valve seat just as easily as one of the maximum size that it is capable of machining. The machine will work with equally satisfactory results in either a horizontal or vertical position.

The spindle and quill revolve together and there are no shoulders or collars to wear and allow the cutter to flinch

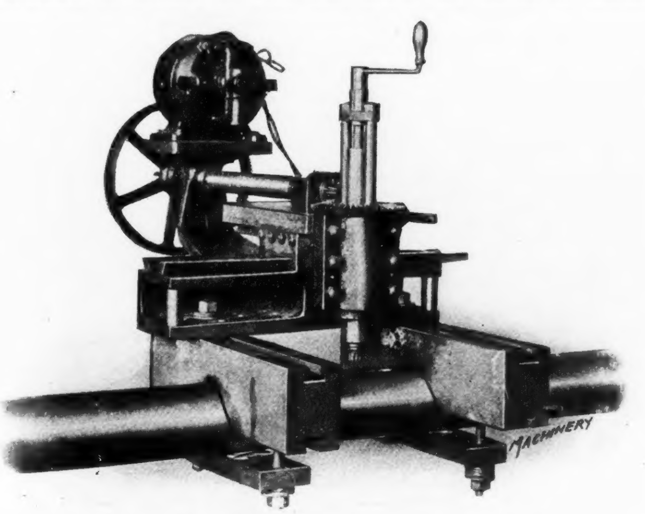


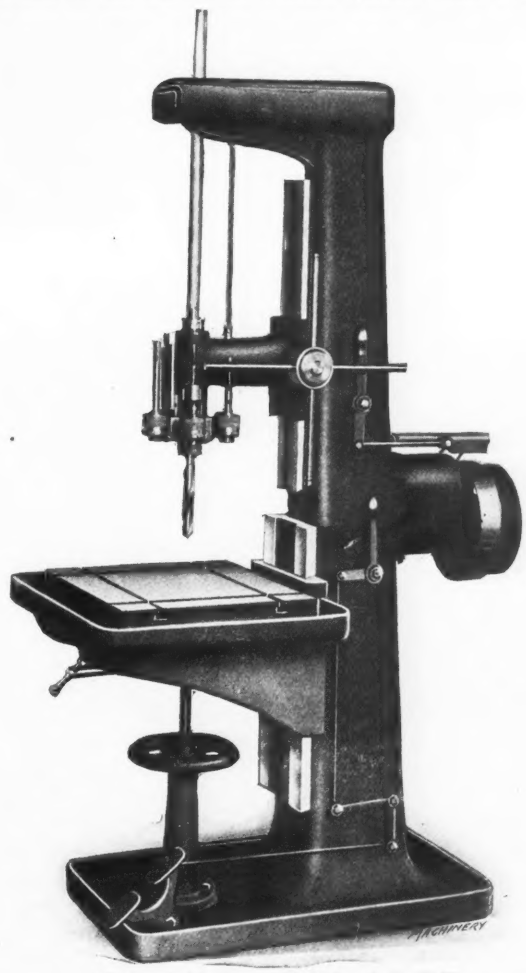
Fig. 2. Pedrick Machine milling a Keyway in a Shaft

away from the work. Another feature is that when the cutter is no larger than the diameter of the spindle, it may be lifted out of the bearing with the spindle to enable the cutter to be sharpened or a new one to be used. This is accomplished by merely loosening two nuts and is found to be a particularly convenient feature of the machine, because if a cutter is working in a small deep chest where there is not sufficient room to reach in and unscrew it, it is not necessary to remove the entire machine. The spindle has a standard taper so that an ordinary cutter may be used.

The telescopic driving shaft works smoothly and quietly throughout its full range, and is of ample strength to meet the requirements of any service for which the machine is intended. The No. 1 machine will face a surface up to 12 by 8 inches in size, and the spindle has a vertical adjustment of 6 inches. The direct motor drive is arranged as shown in the illustrations, and a machine equipped in this way is a self-contained unit that may be used in close quarters and easily set up. Besides using the machine for milling seats in pumps and engines, or pads on large frames and housings, it is found to constitute a very convenient means of machining keyways in shafts. While not particularly designed for the purpose, this machine may also be used for drilling.

ROBBINS MULTIPLE-SPINDLE DRILL

The Taylor quick-change multiple drill press illustrated herewith is a recent product of the Robbins Machine Co., Worcester, Mass. This drill was designed to meet the demand for a machine of the sensitive drill type with sufficient power to handle work beyond the capacity of a sensitive drill press, and at the same time operate quicker than a standard



Taylor Quick-change Multiple-spindle Drill

upright drill. It will drive a 1¼-inch drill as easily as the ordinary sensitive drill press will handle a ¾-inch drill. The feed is by hand, and when large drills are to be used at high speed, compound gearing in the sliding head can be quickly engaged to increase the pressure of the drill and decrease the pressure required on the hand lever. This enables a large drill to be forced to its full capacity by the application of a limited amount of hand pressure. The reverse motion of the spindle is of unusual strength and provides ample power to handle reasonably large taps.

The machine is driven by spiral gears running in oil, and all gears that are subjected to excessive strain are made of heat-treated nickel steel. The quick-change gear-box is arranged for four spindle speeds and one reverse speed for tapping. The spindle speeds are quickly changed without requiring the machine to be stopped. The sliding head affords quick action in raising or lowering the spindles for tools of different lengths.

The quick-change drill chuck is designed to hold five tools such as drills, reamers, counterbores, taps, etc., which are required for performing successive operations on a piece of work. This chuck provides for rapidly moving any tool to the working or idle position without stopping the machine. When the tool to be used is swung to the operating position it is securely locked and held in perfect alignment with the driving spindle. The power is applied through a combination clutch and friction device, and the greater the strain on the tool the more powerful the drive becomes. When the machine is in operation all tools in the chuck are idle with the exception of the working tool. The clutches for holding the tools in the chuck are made to fit either No. 1, 2 or 3 Morse taper shanks.

The machine is ball bearing throughout and is practically noiseless in operation. The principal dimensions are as follows: distance from center of spindle to face of column, 10 inches; maximum distance from table to spindle, 30 inches; traverse of table, 12 inches; traverse of spindle, 16 inches; size of work-table, 18 by 24 inches; diameter of spindle, $1\frac{1}{4}$ inch; and net weight of machine, 1200 pounds.

MURCHEY PIPE THREADING DIE

The accompanying illustrations show a pipe threading die which has recently been placed on the market by the Murchey Machine & Tool Co., Porter and Third Sts., Detroit, Mich. The important feature of the design of this die consists of the four circular chasers which are similar in shape to circular forming tools. These chasers are provided with a series of annular V-grooves properly spaced for the pitch of the thread that it is desired to cut.

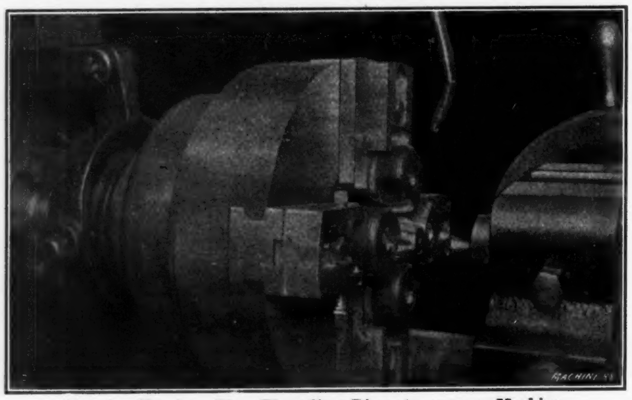


Fig. 1. Murchey Pipe Threading Die set up on a Machine

The circular chasers are carried by slides which fit in T-slots in the die head, the construction being clearly shown in Fig. 1. Interposed between the top faces of these slides and the chasers, there are adjusting plates, each of which has nine holes which are equally spaced. Three pins held in the chasers fit into these holes and adjustment of the position of the chasers is effected by loosening the square screws, lifting the chasers up and turning them around so that the three pins enter another set of holes. The cutting edge of the chasers is then set in the correct relation to the center of the work by adjusting the set-screws in the slides.

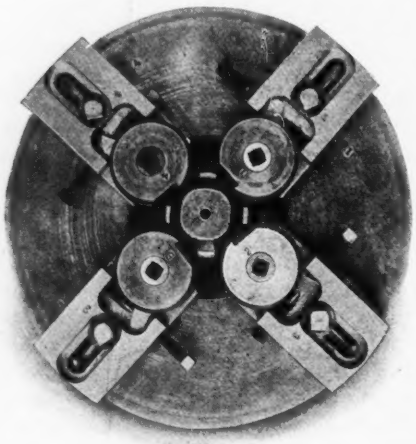


Fig. 2. End View of Murchey Pipe Threading Die

In order to cut the thread on a taper, the top faces of the slides against which the chasers are held are beveled. This serves to throw the chasers to the required angle. Different diameters of pipe are threaded by the substitution of circular blocks of varying lengths in semicircular grooves provided in the top face of the die head and the lower surfaces of the slides. The chasers can be resharpened until practically the entire circumference is ground away so that their life is much longer than that of the ordinary type of die chasers. A reamer is shown in Fig. 2 which is held in the die head to remove the burrs from the end and hole in the pipe.

STOCKBRIDGE KNIFE GRINDER

The time spent in removing and replacing the knives of planers and jointers used in wood-working shops constitutes a large part of the time involved when it is necessary to take them off for grinding. This is a difficult and unsatisfactory method, and is often done hurriedly and improperly, with the result that the machine when started up operates inefficiently and produces a poor grade of work. This is especially true of the thin hard knives used on cylindrical head planers which are almost impossible to grind and reset in perfect alignment for the entire length of the head.

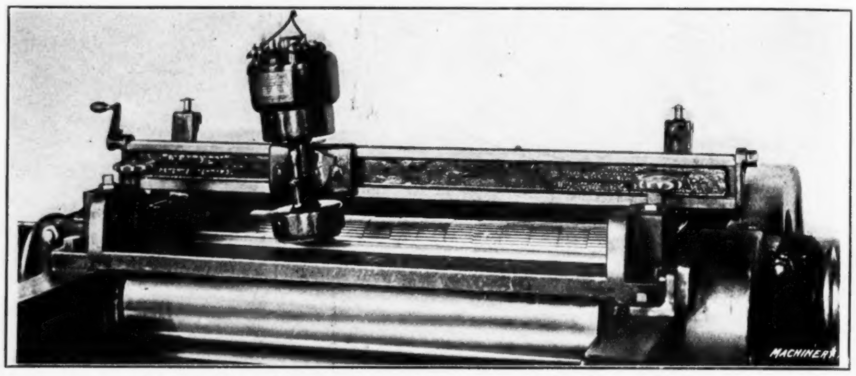


Fig. 1. Use of Stockbridge Grinder in sharpening Planer Knives

In the "Quicsharp" grinder, which is a recent product of the Stockbridge Machine Co., Worcester, Mass., these difficulties are avoided by providing a grinder which enables the knives to be sharpened while in place on the machine. The grinder is motor-driven, the motor being mounted on the head and supplied with a current from an ordinary lamp socket. The saddle has a split nut attached to it which engages the feed-screw located at the top of the bridge. The saddle can be fed across the entire length of the bridge in either direction and at any desired speed. The bridge is supported at the ends or at intermediate points by two angular brackets, which are bolted to the bed of the wood-working machine, and hold the grinder perfectly rigid. The wheel is cup-faced and can be raised or lowered by a thumb-screw to grind the required amount from the knives. The wheel is fed automatically to a positive stop. The grinder head is pivoted at

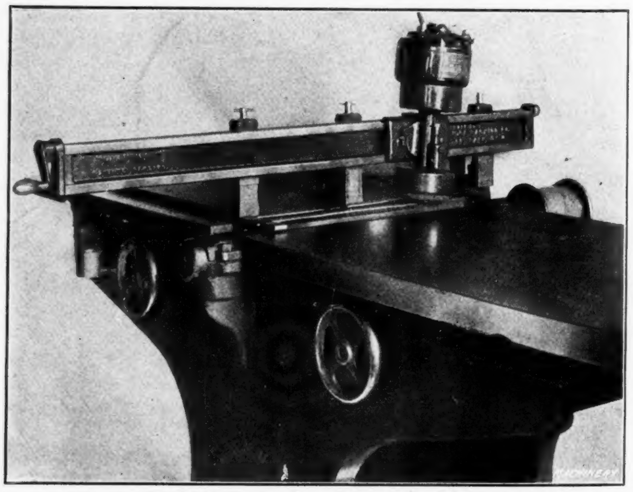


Fig. 2. Stockbridge Grinder sharpening Jointer Knives

the center and can be tilted to either side of the perpendicular; it is held in place against a stop, giving the same angle on each side of the perpendicular. With the head tilted, the knives are ground concave. A positive stop holds each knife in exactly the same relative position to the wheel; therefore, each knife must be ground true. Figs. 1 and 2 show this grinder in use on a planer and on a jointer, and reference to these illustrations will show the way in which it is used, making further description unnecessary.

DERIHON PORTABLE HARDNESS TESTING MACHINE

The illustrations presented in this connection show a portable machine for conducting the Brinell hardness test, which consists of making an impression with a hardened steel ball 10 millimeters in diameter, by the application of a pressure of 3000 kilograms. Fig. 1 shows the machine ready for the test with the lever resting on the shaft. The piece to be tested is placed on the table of the machine, which is then raised until the piece is in contact with the ball. This done, the lever is pulled slowly over so as to give a progressive pressure which is registered by a small manometer until 3000 kilograms is applied. When this figure is reached the lever is slowly returned to its former position and the test is completed. Under normal conditions it is usually sufficient to move the lever through an angle of 45 degrees to obtain the required pressure of 3000 kilograms.

With each machine a small piece of steel is furnished in which a standard impression has been made, the size of the diameter being stamped on it. This standard piece is of

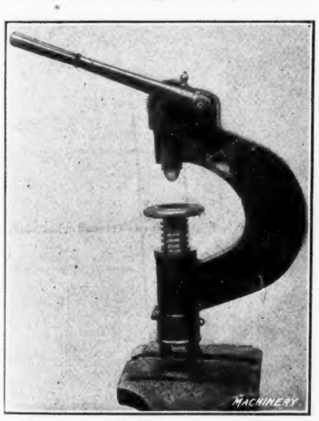


Fig. 1. Machine ready for making Test

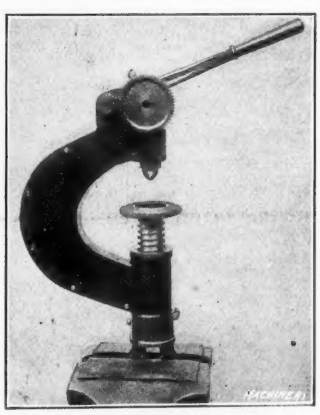


Fig. 2. Arrangement of Gears at Rear of Machine

special air-hardening chrome-nickel steel. The accuracy of the machine can be easily controlled by making an impression beside the standard impression. The construction of the machine is based on the principle of elasticity of the frame, which for this purpose has been constructed in the shape of a horse-shoe. The power produced by the pressure of the ball on the test piece has a tendency to open the frame to a certain degree in proportion to this power. The shape of the frame has, therefore, been specially considered in order to have it as elastic as possible. The pressure exerted of 3000 kilograms does not change the resistance or elasticity of the frame, as it is made of steel having an elastic limit of 242,000 pounds per square inch and a pressure of 3000, 4000 or even 5000 kilograms does not work it above 10 kilograms per square millimeter or 1442 pounds per square inch. Under these conditions, repeated tests even in large numbers do not alter the calibration of the machine.

The deflection of the frame being relatively weak (1 to 1.5 millimeter), a register, the construction of which resembles a metallic manometer, is installed in the hollowed out part of the frame. By means of a needle and a graduated dial, the deflection and therefore the pressure exerted in making the test can be quickly and easily read. To adjust the machine, all that is necessary is to open the head enclosing the mechanism above the frame. Should the machine ever get out of adjustment, a comparison should be made on the standard test piece, and when an impression of the same diameter has been made, the needle is brought over the figure "3000" by means of the small adjusting screw. This adjustment, however, would only be necessary through some accidental cause independent of the operation of the machine under normal usage. Fig. 2 shows the arrangement of the gears at the rear of the machine. This machine is placed on the market by H. A. Elliott, 507 Majestic Bldg., Detroit, Mich.

SCULLY-JONES ADJUSTABLE SPACING COLLAR

Fig. 1 shows the "Wear-ever" adjustable spacing collar for use on milling machine arbors, which is a recent product of Scully-Jones & Co., 349 Railway Exchange Bldg., Chicago, Ill., and Fig. 2 illustrates the method of using these collars. The "Wear-ever" collar was designed primarily for use in connection with manufacturing operations on milling machines, where two or more cutters on an arbor must be spaced at a specified distance from each other. Those familiar with the application of straddle or gang cutters in milling machine operations know that it is sometimes necessary to grind the sides of the teeth. This necessarily changes the distance between the faces of the cutters, and in order to maintain the required dimensions of the work, compensation must be made for the amount which has been removed from the cutters by grinding. This is sometimes done by carrying solid spacing collars of assorted lengths in stock. But if the exact size cannot be found among the collars on hand, it is necessary to grind down a collar that is too long or to shim up one that is too short. While this is being done the milling machine is standing idle.

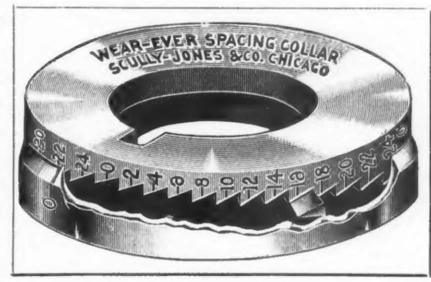


Fig. 1. Scully-Jones Adjustable Spacing Collar

The "Wear-ever" collar is designed so that the thickness may be varied up to 0.024 inch, this adjustment being divided into twelve steps of 0.002 inch each. It will be seen from Fig. 1 that the collar is provided with three sets of graduations ranging from zero to 24 by even numbers, each graduation being opposite a notch in the collar. These notches are arranged on cam surfaces, and corresponding notches in each of the three series of graduations are engaged by a tooth. In making the adjustment, the collar is moved around so that the teeth engage with the proper spaces to give the required expansion of the collar on the arbor. The adjustment is quickly made, and after the collar has been set it is exactly as rigid as a solid collar. The advantages of the "Wear-ever" collar may be briefly summarized as follows:

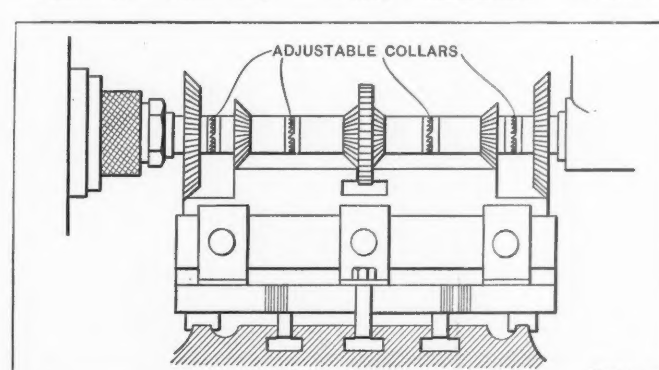


Fig. 2. Use of Scully-Jones Adjustable Spacing Collars

they are made in sizes down to less than ½ inch in width and are capable of standing up under hard service; it is impossible for chips to interfere with the operation of the collar, and the first cost is low so that a shop can afford to purchase them for any work upon which they could be used to advantage.

LOWE "LAST-WORD" INDICATOR

Henry A. Lowe, 1374 E. Eighty-eighth St., Cleveland, Ohio, is now manufacturing the "Last-word" test indicator which is illustrated in Figs. 1 and 2. Fig. 1 shows the indicator carried on the ball joint toolpost shank, and this illustration also shows an auxiliary clamp which is provided with the instrument. This clamp enables an instrument to be removed from the toolpost shank and set up on the needle of a surface gage as illustrated in Fig. 2. There is ball joint connection between the toolpost shank and the indicator, which provides adjustment to meet the requirements of a great variety of work.

The contact lever has a hardened taper-head stud for a bearing, which provides adjustment for wear. There is a small tapered hole through the contact ball, and contact points of special shape can be fitted into this hole to meet the requirements of special classes of work. In many cases the ordinary contact ball is quite satisfactory without providing any auxiliary point.

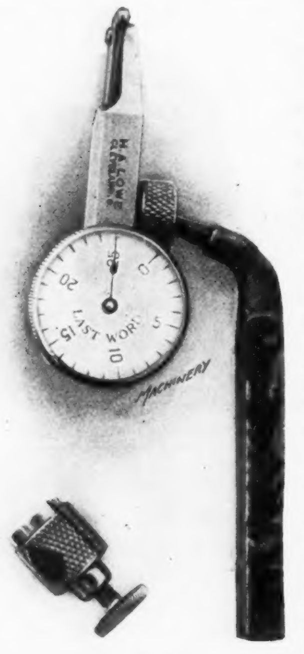


Fig. 1. Lowe Test Indicator carried on Ball-joint Toolpost Shank

The mechanism of this indicator has been worked out to give the magnifying power of 100, and at the same time the instrument is of remarkably small size, which will be appreciated when it is known that the weight is only 1¼

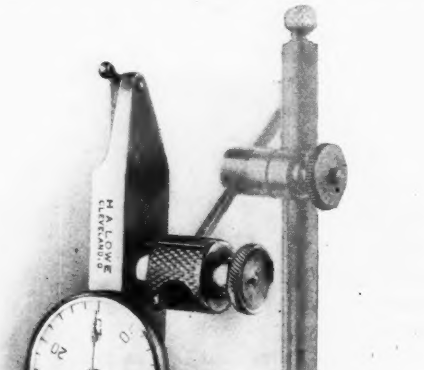
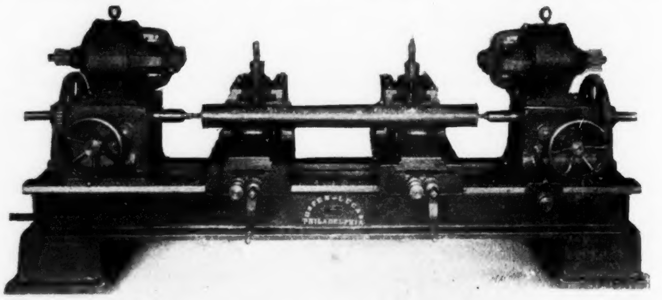


Fig. 2. Lowe Indicator mounted on the Needle of a Surface Gage

ounce. The dial is graduated to read in 0.001 inch. This instrument can be quickly changed from the toolpost shank to the needle of a surface gage to adapt it for surface plate work, or back to the toolpost shank for lathe, shaper, planer, grinding or milling-machine work. It is adaptable for use on a great variety of tool-room and machine-shop operations.

ESPEN-LUCAS AXLE CENTERING MACHINE

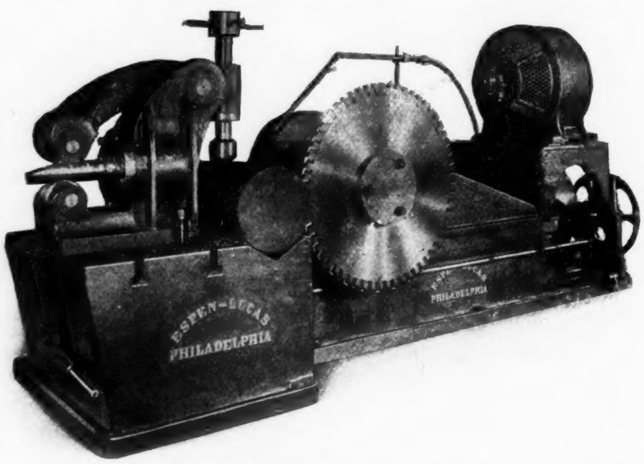
The Epsen-Lucas Machine Works, Front & Girard Aves., Philadelphia, Pa., has recently placed a double-end axle centering machine upon the market. This machine is intended for centering both ends of axles up to 7 inches in diameter by 8 feet long. The machine is arranged with two


Espen-Lucas Double End Axle Centering Machine

spindles which are driven independently by 2-horsepower motors. On head is stationary and the other adjustable for centering axles from 5 to 8 feet in length. The two cradles are provided with rollers for centering the axles and heavy clamps for holding the work rigidly in place. The weight of the machine is about 11,000 pounds.

ESPEN-LUCAS COLD SAW

The accompanying illustration shows a heavy type of cold saw which is one of the recent products of the Espen-Lucas Machine Works, Front & Girard Aves., Philadelphia, Pa. This machine is equipped with a saw blade 60 inches in diameter and has a capacity for sawing round or square bars up to 20 inches thick. The feed and speed variations are so arranged that the machine can be used for sawing at an advance of 6 inches per minute. The drive is through a main shaft, a phosphor-bronze worm-wheel and a crucible steel worm running in oil. The spindle extends for the entire width of



Espen-Lucas Cold Saw Machine

the carriage and the saw blade is bolted directly against the driving gear, an arrangement which practically eliminates torsional strain between the driving gear and the saw blade.

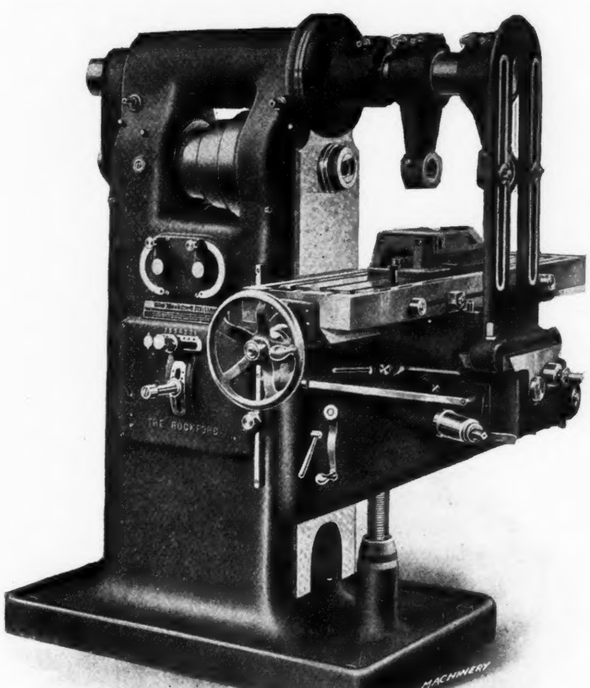
The feed is obtained from a geared friction plate controlled by an automatic lock-nut lever, by which any change in the advance of the saw can be made while the machine is in operation. The saw carriage also has a quick return movement controlled by the same lever. An automatic stop for controlling the traverse of the carriage is also provided. Any type of saw blade including the inserted tooth or solid high-speed steel blades may be used.

Compressed air or screw clamps are provided for use on this machine, depending upon the requirements of individual cases. For handling stock rapidly where multiple cuts are to be made, this saw is equipped with a stock feed attachment which will handle bars of any length up to 20 feet. The usual form of drip pan furnished on machines used for sawing smaller sizes of stock has been eliminated in the

present instance and a reservoir substituted in the foundation. The machine is belt-driven from a 35-horsepower motor and weighs 50,000 pounds approximately.

ROCKFORD HEAVY PLAIN MILLER

The Rockford Milling Machine Co., Rockford, Ill., is now manufacturing the No. 2 heavy double back geared cone type plain miller which is illustrated herewith. This machine is of exceptionally heavy construction, the approximate weight being 3750 pounds. The back gears are enclosed in the column and operated by two levers which afford a very convenient



Rockford No. 2 Double Back Geared Plain Miller

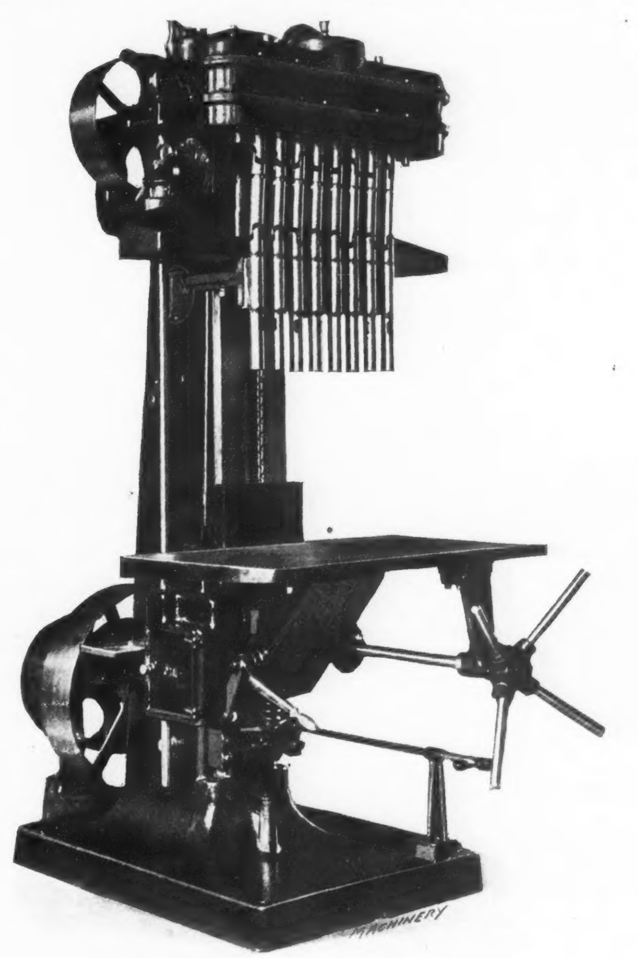
and practical control. The machine is equipped with a flanged support for the overhanging arm, which has been designed by this company; this support stiffens the construction and increases the cutting capacity of the machine. The diameter of the overhanging arm is 4 inches.

The table is provided with a quick return movement operated by a handwheel and clutch, and the handwheel can be thrown in instantly at any desired point. A wrench can be used on the squared shaft at the opposite end of the table. The table measures 50 by 11½ inches over all, and is provided with three ⅝-inch T-slots; there are also oil grooves on the sides and oil pockets at the ends of the table. All feeds are automatic. The spindle is bored No. 11 B. & S. taper, and is made of crucible steel; it runs in phosphor-bronze bushings which have tapered journals fitted with oil retainers. Wear is compensated for at the rear of the spindle by means of a nut. There are eighteen speed changes ranging from 17 to 392 revolutions per minute, and fourteen changes of feed ranging from 0.005 to 0.175 inch per revolution. Feed reverse levers controlling all feeds are located on each side of the knee. The regular equipment of the machine consists of a plain vise, a flanged support for the overhanging arm, two arbor supports and the necessary wrenches. The design has been carefully worked out to secure the maximum rigidity.

FOX MULTIPLE-SPINDLE RAIL DRILL

The No. 3 Fox multiple-spindle rail drill illustrated in this connection has been developed to meet the demand of engine builders for a high-speed multiple-spindle machine for drilling and boring push-rod holes, valve stem guide holes and cages. Particular attention has been paid to the design of the vertical adjustment for each spindle with the view of keeping the cutting edge of all tools in the same plane. This is a feature which users of the machine will appreciate. The base, column and gears for changing the speeds and table feeds are of the same construction which has been used in the past on Fox multiple-spindle drilling machines. The gears are cut from solid 35-point carbon steel blanks, which are pack-hardened and heat-treated. The bearings for the gears are subsequently ground, giving the most perfect construction possible. The principal bearings in the machine are bronze bushed, and the main driving pulleys run on Hyatt roller bearings.

Rails of various lengths are furnished with this machine to meet the requirements of individual cases. The rails are made with either a tongue which fits grooves in the adjusting arms to keep all spindles in positive alignment, or the tongue may be omitted from the rail, giving universal adjustment to each spindle. The rail is an exceptionally heavy casting ribbed to provide the necessary rigidity. The gear case containing the driving pinions for the spindles provides a double bearing for each gear in the case. The gears are of 6 and 8 pitch, with broad faces. The regular gear case is bored for a maximum of twelve spindles and when less than this are furnished with the tool, the extra holes are plugged up; it is thus possible to add additional spindles should they be required at any time. The drill spindles, spindle bearings and adjusting arms are of patented con-



Fox Multiple-spindle Rail Drill

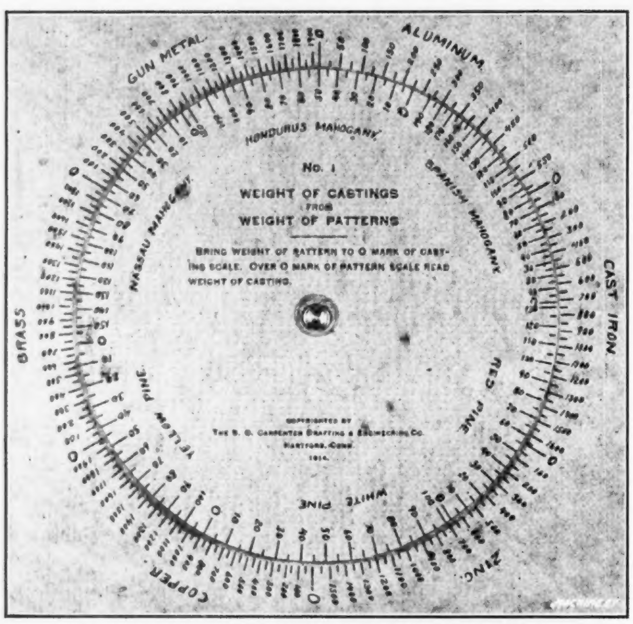
struction, which gives a wide range of vertical adjustment. This adjustment is obtained without changing the set position of the arm. The spindle bearings have a T-slot milled for their entire length and fit into the slotted end of the adjusting arms. T-bolts working in the slots in the bearings secure them to the adjusting arms and provide for adjusting the spindles vertically.

The universal joints used in these machines are made with every friction surface hardened. An oil pocket is provided in the center of the block, giving perfect lubrication. The table regularly furnished on this machine has a surface 24 by 38 inches in size and is equipped with a hand pilot wheel

for raising or lowering. A single lever is provided for manipulating the power feed; pulling up on this lever engages the feed and pushing down on it disengages the feed. An automatic trip lever is provided for automatically throwing out the feed. There are six changes of feed ranging from 0.5 to 5.0 inches per minute; four changes of drill speed are available, depending on the size of drills used. The floor space occupied by this machine is 6 feet by 2 feet 6 inches; the height is 7 feet 8 inches; and the net weight of the machine is 3288 pounds. This machine is a recent product of the Fox Machine Co., 16 Front St., N. W., Grand Rapids, Mich.

CARPENTER CALCULATING CHARTS

The accompanying illustration shows one of a series of eight calculating charts which have recently been brought out by the S. C. Carpenter Drafting & Engineering Co., 49 Oakland Terrace, Hartford, Conn. It will be seen that this chart is intended for finding the weight of castings of different metals from the weight of the patterns from which they are made, the chart being laid out for a number of different kinds of wood commonly used in pattern making. Other charts of this series which are likely to be of interest



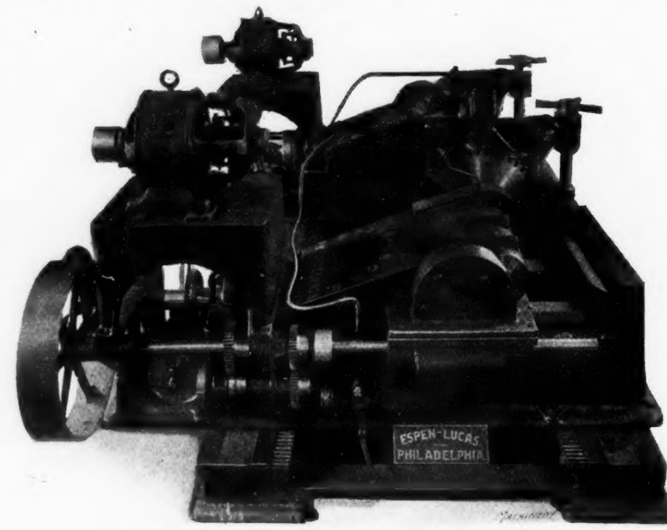
Carpenter Chart for calculating Weight of Castings from Weight of Patterns

to readers of MACHINERY are as follows: Chart No. 2, which gives areas and circumferences of circles with diameters varying by 1/16 inch, and the squares, cubes, square roots and cube roots for each 1/16 inch up to 1 foot; chart No. 3, which gives drill, drill rod, wire and machine screw sizes; and chart No. 4, which gives the sides, angles and area of right angle triangles up to 10 inches base.

ESPEN-LUCAS DUPLEX COLD SAW

The duplex cold saw illustrated herewith is a recent product of the Espen-Lucas Machine Works, Front & Girard Aves., Philadelphia, Pa. This machine is intended for cutting off both ends of axles at the same time, and for this purpose it is equipped with two 32-inch inserted tooth saw blades. The machine has a capacity for sawing off both ends of from twelve to fifteen 9-inch axles per hour. The speed and feed variations provide for sawing at an advance of 6 inches per minute. The cutting capacity is only limited by the nature of the steel that is being cut and the endurance of the high-speed steel cutters in the saw blades.

The machine is driven through a train of compound gearing, a steel worm and a bronze worm-wheel. The gearing and other parts of the machine are liberally proportioned to provide ample strength. The main bed and table are held together by heavy webs and the saw carriages have bearings of ample size on the base of the machine. The saw blades are bolted directly to the driving gears which are forged integral with the ends of the spindles. This arrangement practically eliminates the torque between the driving gear and the saw and enables the spindle bearings to extend for the entire width of the saw carriages. This drive is a strong feature of the machine. All of the gears, spindles and shafts, etc., are cut and turned from high carbon hammered steel forgings. All bearings are bushed with bronze and the thrust



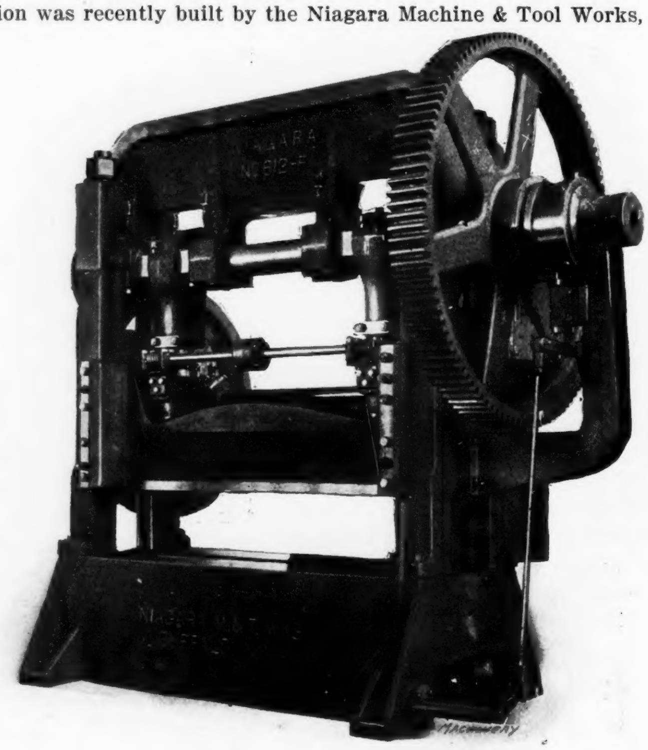
Espen-Lucas Duplex Cold Saw

of the driving worm and other revolving parts is taken up by roller thrust bearings.

The feed is variable and automatic, being controlled by an automatic stop which regulates the depth of cut up to the full capacity of the machine. The saw carriage is equipped with automatic power return and an automatic starting device which leaves the operator free to give his whole attention to the feeding of axles to the machine and removing them when finished. Pumps are provided to furnish a liberal flow of cutting compound to the saws. The machine is driven by 15-horsepower electric motors and weighs approximately 30,000 pounds.

NIAGARA DOUBLE CRANK PRESS

The large double crank press illustrated in this connection was recently built by the Niagara Machine & Tool Works,



Large Niagara Double Crank Press

Buffalo, N. Y. This machine is intended for punching and forming operations met with in steel car work, but the ma-

chine is also suitable for a wide range of heavy stamping operations. The bed, housings and arch are held together by means of four steel tie-rods 4½ inches in diameter which are a shrink fit in the parts which they connect. The same convenient method of adjusting the slide and gibs is provided that is employed on other large presses of this company's manufacture. The motion of the press is controlled by a hand lever operating a powerful jaw clutch. The working surfaces of the three jaws of the gear hub—as well as the clutch collar—are faced with hardened tool steel pieces. An automatic device is provided to stop the slide at the highest point of its stroke. Presses of this type are built in different sizes; the machine shown in the illustration measures 84 inches between the housings and has an approximate weight of 50,000 pounds.

* * *

STATUS OF THE PFAUTER PATENTS IN THE UNITED STATES

Referring to the note appearing on page 478 of February MACHINERY, engineering edition, in reference to the Pfauter patents on gear-hobbing machinery, we wish to state that this seems to have been misleading, some of our competitors assuming that they are now in a position to offer gear-hobbing machines with the differential for cutting spiral gears in the American market also. In order to correct this misunderstanding, we call attention to the fact that the United States patent which was granted to Herman R. Pfauter under date of January 2, 1900, has still about three years to run. Anyone selling or offering for sale gear-hobbing machines equipped with differential gearing for cutting spiral gears will infringe the existing Pfauter patents, and Mr. Pfauter is fully prepared to prosecute any such infringement.

New York City SCHUCHARDT & SCHUTTE

* * *

NEW MACHINERY AND TOOLS NOTES

Heating Forge: Word Brothers, San Francisco, Cal. A forge heated by oil which was particularly designed for heating rock drills; this forge is also adapted for general blacksmith work.

Roller Bearing: Royersford Foundry & Machine Co., 54 N. 5th St., Philadelphia, Pa. A new type of Sells roller bearing in which the sleeve is split and clamped over the shaft by two split collars.

Finishing Tool: Western Tool & Mfg. Co., Springfield, Ohio. A tool for finishing work in the lathe, shaper or planer which consists of a forged shank to which a steel disk which constitutes the cutter is bolted.

Drill Head: Edward Board, 619 Filbert St., Philadelphia, Pa. A four-spindle drill head designed for drilling four holes simultaneously at an angle of 45 degrees. This head is particularly designed for machining the burners of gas stoves.

Time Recorders: Baird Equipment Co., Chicago, Ill. Three types of time recorders intended for use in marking employe's weekly cards; daily job tickets; single, three or seven day job cards, or for use as a watchman's time detector.

Magnetic Chuck: O. S. Walker & Co., Worcester, Mass. A multiple pole magnetic chuck which is similar in general construction to previous types of magnetic chucks of this company's manufacture. Each pole is energized by an individual coil.

Punch and Shear: J. W. Grace Co., Burlington, Vt. A hand operated punch and shear equipped with two sets of blades. This shear will handle flat bars up to 3 by ½ inch in size. The punch has a capacity up to a ½-inch hole pierced through a ½-inch iron plate.

Facing Arm: H. B. Underwood & Co., 1024 Hamilton St., Philadelphia, Pa. A universal swiveling facing arm on which the tool-holder swivels through a complete circle and can be clamped in any position. The facing arm can be applied to either end of the bar.

Tool Racks: The New Britain Machine Co., 64 Bigelow St., New Britain, Conn. A line of steel tool racks with seamless sheet metal trays pressed from No. 16 gage steel. Racks of this type are mounted on wheels where it is desired to provide for moving them from place to place.

Circular Slide Rule: Lucien E. Picolet, 19 S. 9th St., Philadelphia, Pa. A circular type of slide rule particularly intended for performing the operations of multiplication, division and the extraction of square and cube roots. The results obtained with this rule are claimed to be as accurate as those obtained with three place logarithms.

Brazing Compound: Phillips-Laffitte Co., Philadelphia, Pa. A compound known an "Unifonte" for brazing cast iron. With this material any mechanic can braze broken cast-iron parts and produce a joint which is claimed to be stronger than the original material. "Unifonte" consists of a chemical paste which is used in connection with flux and spelter.

Pneumatic Drill: Ingersoll-Rand Co., New York City. The improved "Little David" pneumatic drill of this company's manufacture is equipped with roller bearings for the connecting rods and crankshafts which run in ball bearings. In other respects the drill is of similar design to the preceding model which has been manufactured by this company.

Cutting-off Machine: Grant Engineering Co., Detroit, Mich. A machine with a capacity for cutting stock up to 4 inches in diameter. The work remains stationary and is cut by a revolving cutter head which is worm driven. The speed of the head is automatically accelerated as the tools travel toward the center, thus maintaining a constant peripheral speed.

Thread Rolling Machine: The Waterbury Farrel Foundry & Machine Co., Waterbury, Conn. A reciprocating machine for rolling threads on very small screws. This machine has automatic feed, and the capacity is for blanks up to an approximate diameter of 1/16 inch. The feed mechanism is so designed that short work can be handled to excellent advantage.

Belt Pole: Edward Wilbur, 125 Summer St., Boston, Mass. A belt shifting pole designed to eliminate the use of ladders in shifting belts, and at the same time to facilitate the rapidity with which shifts can be made. The pole is 9 feet long, and has a fork at the upper end, one prong of which passes under the belt while the other acts as a fulcrum on the edge of the pulley.

Boring Mill: Cincinnati Planer Co., Cincinnati, Ohio. A boring mill of lighter construction than the machines previously built by the company. The features of the present machine are centralized control, complete guarding of all dangerous parts of the mechanism, a housing of increased rigidity, and double bearings for the table pinion. Machines of this type are built in 6, 7 and 8 foot sizes.

Planer Type Surface Grinder: Newton Machine Tool Works, Inc., Philadelphia, Pa. A line of heavy surface grinders intended for grinding locomotive radius links, tongue pins, armor plate, etc. The abrasive wheel remains in a fixed position while the work held in pneumatic chucks is traversed back and forth on the table. The wheel is adjusted for the full depth of cut and full width in one operation.

High-speed Hacksaw: Armstrong-Blum Mfg. Co., 343 N. Francisco Ave., Chicago, Ill. A high-speed hacksaw similar in design to the machine described in the June, 1911, number of MACHINERY. The present machine, however, is not provided with the cross-feed movement for advancing stock ready for a subsequent cut after a piece has been cut off. The capacity of the machine is for 6-inch square bars.

Plain Grinding Machine: Modern Tool Co., Second and State Sts., Erie, Pa. A small size grinder for manufacturing purposes. In designing this machine particular attention has been paid to features which will lessen the cost of production of work which requires grinding. The machine is known as a No. 8 size, and has the same capacity as the No. 1 universal grinding machine of this company's manufacture.

Self-opening Die: George Overton, 821 E. 167th St., New York City. A self-opening die of simple construction which has four chasers carried in blocks. In operation, the die travels over the work until the turret slide comes against its stop, when the die is opened through inertia. If preferred, an internal stop in the die shank can be set so that the end of the work will strike it and cause the die to be opened.

Two-speed Hacksaw: W. Robertson Foundry & Machine Co., Buffalo, N. Y. This company is now equipping its hacksaw machines with a two-speed mechanism designed to give suitable speeds for cutting tool and soft steels. In this way a maximum amount of service is obtained from the blades. The two speeds are obtained through a set of screws consisting of a twin gear mounted on a crankshaft which meshes with either of two pinions on the driving shaft.

Vertical Lathe: Bullard Machine Tool Co., Bridgeport, Conn. A 6-spindle vertical automatic lathe of the station type. The machine is equipped with independent work-tables, independent tool heads, cross feeds for all boring and turning cuts and a positive maximum feeding pressure for each tool head. All dangerous parts of the mechanism are adequately guarded and positive stops are provided for the tool head movements which work to 0.001 inch. A retarding mechanism is provided for indexing the spindle carrier and a controller interlocks all of the automatic movements. The

time of machining a piece on this vertical lathe is the time of the longest operation plus the time of one indexing. The weight of the machine is 16,000 pounds.

* * *

CENTRALIZED CONTROL RADIAL DRILLING MACHINE

Centralized control of machine tools has received a growing amount of attention of late years. High cutting efficiency is another equally important requirement. It has been the aim of Messrs. J. Archdale & Co., Ltd., Ledsam St., Birmingham, England, in the machine here illustrated and described, to develop these two features to the highest degree.

In the radial drilling machine illustrated in Fig. 1, centralized control has been achieved by the bold step of embodying all the speed changes in the saddle itself. The changes of speed are made by two levers operating steel sliding gears, a plate giving lever positions for the various rates of speed and feed being fixed to the saddle close to the speed change levers. It will be obvious that speed control is quicker and easier than on machines where the speed changes are made in a gear-box carried on the baseplate at the back of the column. The reversing gear is also carried on the saddle, and is operated by a convenient lever. Thus all speed changes can be made, and the spindle started, stopped or reversed without the operator changing his position. The fact that all transmission shafts and gears in this design run at a constant high speed, results in a greatly increased efficiency.

Another important feature bearing on the question of efficiency is the automatic lubrication of all gears, shafts, etc., in the saddle, which reduces the time and attention required for oiling to a minimum. Gear-driven feed motion is provided with changes made by lever, the friction clutch being operated by a lever which is always in the same position. An instantly adjustable automatic stop is provided. The spindle runs in phosphor-bronze bearings carried in a steel sleeve, is provided with a ball thrust bearing, and balanced by a compensated spring arrangement.

Naturally, the complete saddle is heavier than an ordinary saddle, but the rollers on which it is carried enable the movement along the arm to be made with the greatest ease. The traversing wheel is shown at *F*, Fig. 2, which shows a view of the saddle mechanism with the front cover removed.

Fig. 1. Archdale Radial Drilling Machine

The mechanism for sliding the spindle gears *A* and the back gears is clearly indicated, *J* and *H* being two levers operating the sliding gears through the helical pinions *C* and racks *B*.

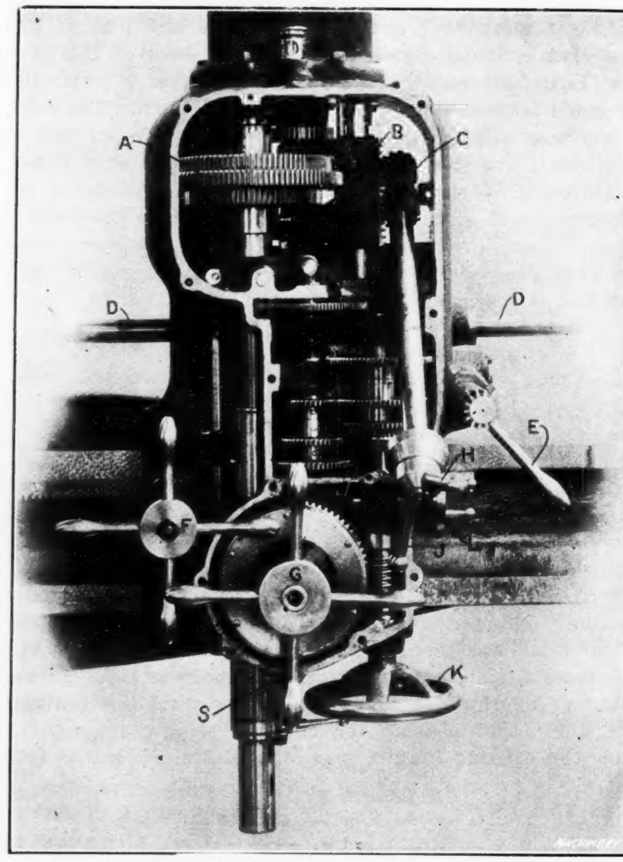


Fig. 2. View of Saddle Mechanism with Front Cover removed

The feed control is obtained at *L*. Quick and slow handfeed motions are obtained by handwheels *G* and *K*. The reversing lever is shown at *E*.

The arm is of an oval box section, and it is claimed that this is superior to the ordinary composite section of a box with ribs top and bottom. Severe tests have shown that torsional deflection is so minute as to be practically negligible. The fixed column carries a rotating sleeve on which the arm is vertically adjustable, this sleeve being mounted on roller journal bearings and a ball thrust bearing, making the swinging of the arm extremely easy. A convenient lever is provided for rigidly clamping it to the fixed column. A friction gear-box at the top of the sleeve operated by a convenient handle controls the vertical movement of the arm, safety stops being provided to prevent over-running.

The base is double-ended and has an extension at the back. Thus, as drilling proceeds on one portion, work may be set up at either one or both of the two remaining positions. The operator, therefore, can be drilling practically the whole of the time, moving from one position to another. As setting up of work on drilling machines takes up a considerable portion of the whole time, it is clear that this is a valuable feature where a large output is required. The box table may be used in either of three positions or swung clear of the base. The motor, which rotates with the arm and sleeve, is at the top of the column, and therefore can never be an obstruction to work being set up on the baseplate. Where the work is such as requires lubricating, the pipe is taken up through the center of the column, fitted with swivel joint, and connected by flexible pipe to the saddle. The lubricant is thus available for all positions of the spindle or arm with-

out danger of fouling the pipe. The pump is, in this case, a separate unit, being connected to the machine by pipes, preferably under ground, in covered troughs.

* * *

AIR BRAKE TESTS, PENNSYLVANIA RAILROAD

Important improvements in the braking of heavy passenger cars were described in a paper read before the American Society of Mechanical Engineers at the Engineering Building, New York City, February 10, by S. W. Dudley of Pittsburg. Air brake tests were conducted jointly by the Pennsylvania R. R. and the Westinghouse Air Brake Co., and the results are considered the most important recent contribution to the subject. A train of twelve steel cars running at sixty miles per hour stores up 224,000,000 foot-pounds of energy. This is sufficient to raise the entire train 120 feet. With prevailing brake equipment such a train would be stopped by an emergency application in a distance of 1600 to 2200 feet, according to the truck rigging and brake shoe design. These tests showed that this distance has actually been reduced to 1000 feet or to within the length of the train. This was the result of improvements in the truck brake design involving the clasp brake, having two shoes per wheel, and the location of the brake shoes with reference to the horizontal center line of the wheels, in addition to improved methods of applying the air brakes quickly and simultaneously and at a high pressure. This concerns safety. These tests emphasized, as has never been done before, the possibilities of improvement in efficiency and economy in regular service operation by proper attention to design and installation in order to permit the realization of the flexibility of improved air brake apparatus. These improvements center in the electric control of the brakes, giving quick, simultaneous and responsive action. The electric control has opened the way for maximum effect in practice of improvements in practically all the factors involved in air brake apparatus, all of which were covered in the development represented by these tests. The tests constituted a progressive development of brake rigging and brake shoes in connection with the scientific study of the air brake as a whole. Fifteen years ago trains were stopped in about half the distances prevailing in the practice of today. Increased size and weight of equipment brought an entirely new brake problem. These tests showed that high-speed braking on the longest passenger trains can be accomplished with safety but at the expense of a somewhat complicated apparatus which responds to both pneumatic and electric impulses.

* * *

CLEVELAND AUTOMATIC FOR MACHINING BRASS NUTS

The Cleveland Automatic Machine Co., Cleveland, Ohio, recently installed two of its 7/8 to 1 1/4 inch model C "automatics" in the plant of the Metric Metal Works, Erie, Pa. These two machines are equipped with an air chucking device, a rotary tilting magazine, and special tools to adapt them for facing and tapping brass coupling nuts ranging from a very small size up to 2 1/2 inches in diameter. The machines have now been in operation long enough to show that they are exceptionally well adapted for nut work. The output is three times that which has ever been obtained by any other method.

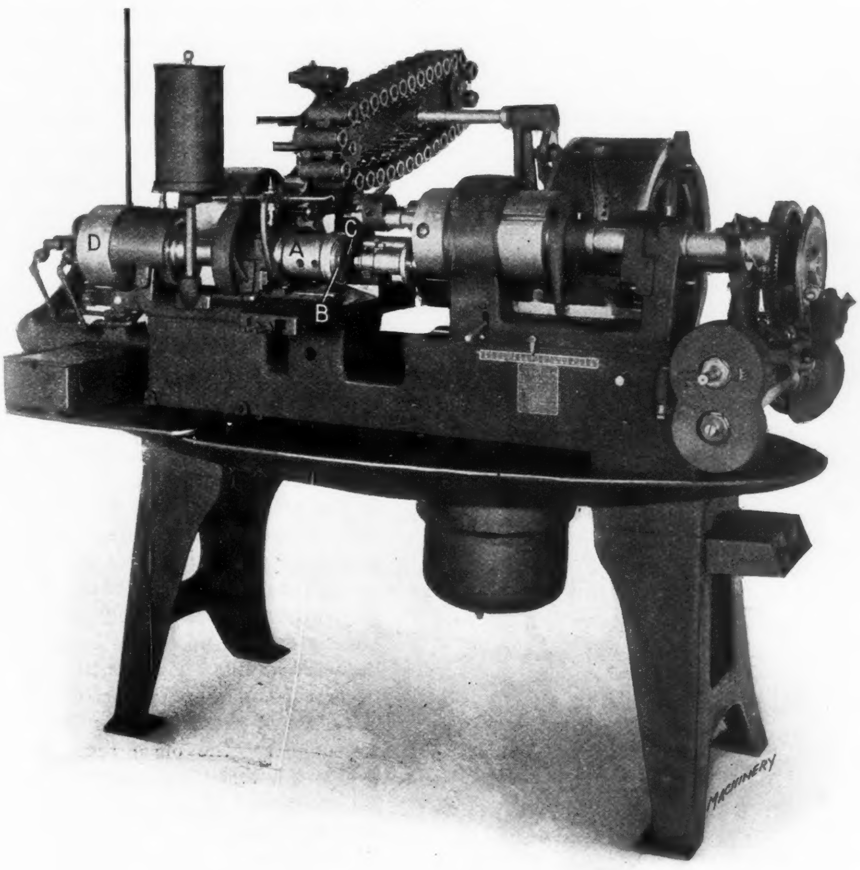


Fig. 1. Front View of Cleveland Automatic for machining Brass Nuts

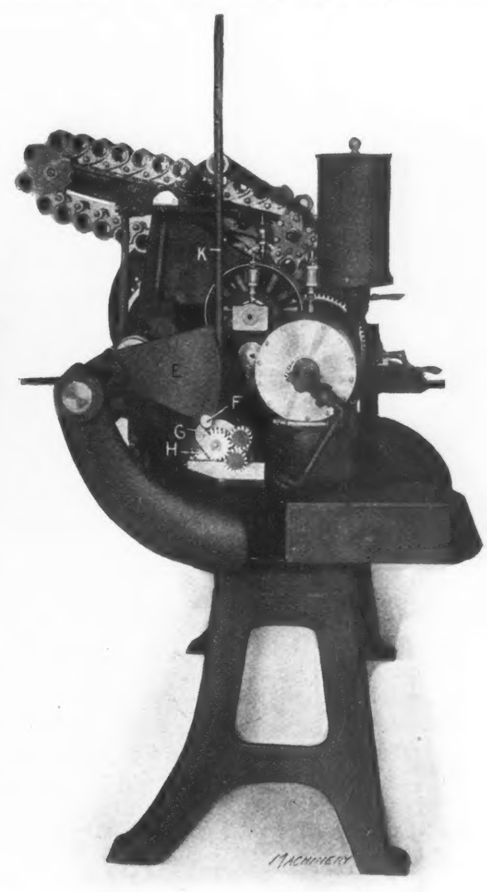


Fig. 2. End View of Machine showing Mechanism for controlling Air Chuck

The air chuck on the machine in Fig. 1 is shown at *A*. The chuck is screwed onto the spindle in place of the regular chuck hood and is fitted with three removable jaws *B* which, in turn, are provided with three pads *C* that can be shaped to suit various kinds of work. The rear end of the spindle is shortened and the air cylinder *D* is screwed onto it. A connecting-rod actuated by the piston in the cylinder *D* operates the chuck jaws *B*, the admission of air at each side of the piston opening and closing the chuck.

Fig. 2 shows an end view of the machine where the mechanism for controlling the admission of air to the cylinder *D* is clearly illustrated. The segment *E* is fitted with a roller *F* in contact with the lever *G* that operates the air valve to release the chuck. The geared lever *H* swings up as the lever *G* is carried down by the roller *F;* a second roller on the opposite side of the segment *E* then comes into contact with the lever *H* as the segment swings around and this moves the valve to its original position, closing the chuck and

at the same time exhausting the air on the opposite side of the piston. The air supply pipe K is fitted with a sight-feed lubricator (not shown) which thoroughly oils the piston and moving parts of the air valve.

The rotary tilting magazine L, Fig. 3, is equipped with a link belt M, each link being fitted with a bushing of suitable shape to hold the work. When the magazine L tilts up after the conveyor N has removed a blank, the lever P comes in contact with the pin R which indexes the link belt to bring another piece of work in line with the conveyor when the magazine swings down the next time. This rotary tilting magazine was described in detail in the February number of MACHINERY, so that further discussion of this feature of the machines which form the subject of the present article is unnecessary.

The conveyor N is fitted with a flanged sleeve S that comes in contact with a stop on the turret head (not shown) on the backward stroke of the turret after the chucking operation. Should the chuck fail to grip the work, the sleeve S strips the work off the conveyor, thus allowing the conveyor to remove another piece of work from the magazine without interference. If the finished work is not ejected from the chuck, the forward movement of the next blank carried by the conveyor comes in contact with it and compresses a spring in the shank of the conveyor N; this takes up the strain and avoids damaging the machine.

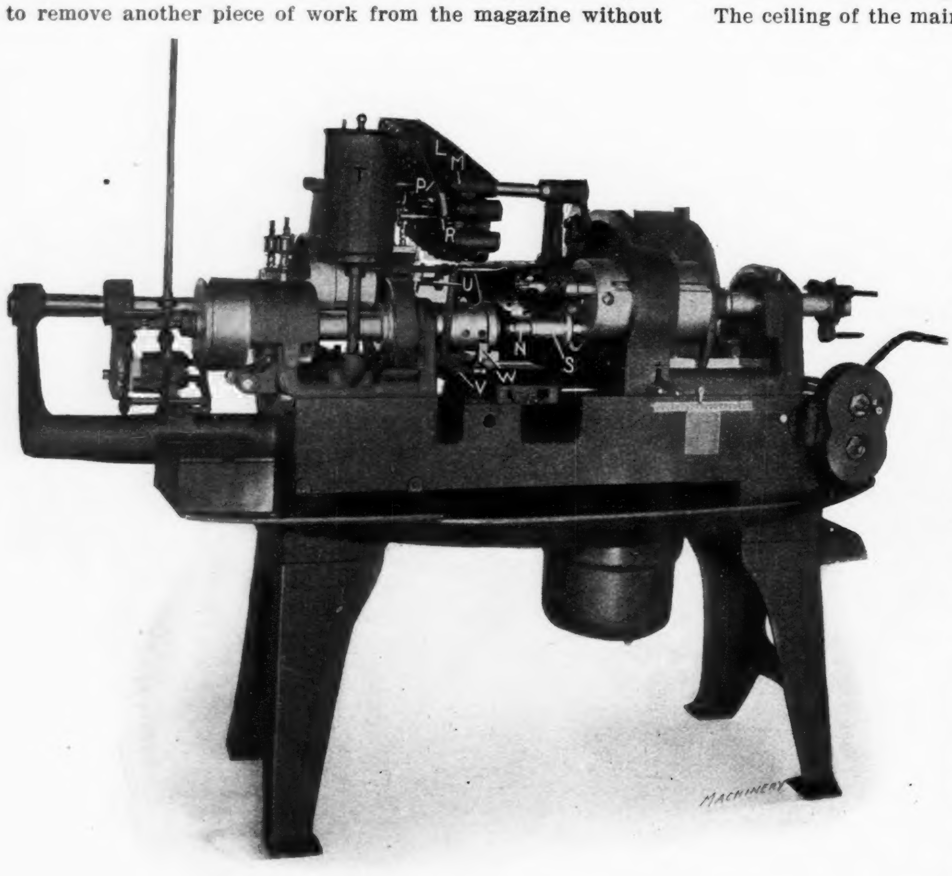


Fig. 3. Arrangement of the Magazine and Chuck Mechanisms

Fig. 3 also shows an oiling device used to drop oil onto the tap. This lubricating equipment consists of the tank T with suitable piping and a valve U which is operated by a lever V pivoted on the spindle head and operated by the tripping dog W on the cross-slide. As the cross-slide moves forward to allow the facing tool on the rear of the slide to operate, the tripping dog W strikes a V-block on the lever V, causing the valve U to open and drop the required amount of oil onto the tap. The amount of oil supplied may be varied from one drop to a steady stream, being under control of the operator.

It has already been stated that these machines were built to meet the special requirements of the Metric Metal Works, but "Cleveland" automatics equipped with this type of air chuck can be used for all sorts of work requiring drilling, counterboring, reaming, tapping, recessing and similar operations.

ADMINISTRATION BUILDING OF NEW DEPARTURE MFG. CO.

The New Departure Mfg. Co., Bristol, Conn., recently moved into its new Administration building, which has been in course of construction for the last twelve months. The demand for New Departure ball bearings has increased so rapidly that the company could not wait to build the required additions to its manufacturing plant, and as a result bought the factory of the Whitlock Coil Pipe Co., Hartford, Conn., which comprises 145,000 square feet of floor space. The office quarters were also inadequate and this deficiency has been met by the erection of the new Administration building, which is 62 feet wide by 220 feet long and six stories high. The building is modern in every detail of construction and is absolutely fireproof. The outer walls are of light grey brick in three shades, relieved by broad ribbons of blue and green colored tiles at the second and fifth stories. The entrance is reached by a broad flight of granite steps, and is flanked on each side by hand-carved limestone panels.

The ceiling of the main vestible of the building is decorated in Roman gold, and the side walls are panelled with marble. The floor is also of marble with a geometrical design worked out in tiles, and with the monogram of the company in green and white as a center piece. The main lobby, which is just beyond the vestibule, is 16 feet square; the floor is of Italian marble and the wainscot is relieved by ornamental capitals. The office is located on the fifth floor of the building and is reached by an electric elevator which runs from the lobby.

The private offices of the officials and heads of departments are located at the front and south side of the fifth floor. The workers in each department are directly opposite the private office of the head of that department. These offices are finished with panels of mahogany carried up to the height of the window sills. The partitions between the general offices are of plate glass above the wainscot. At the west side of the office there is a double fire- and burglar-proof vault. All equipment of this vault is steel and absolutely fire-proof. One of the features of the main office is a large leaded glass dome ceiling light 112 feet in length by 16 feet wide. In addition to the general offices there is a large foremen's conference room on this floor, which has a small kitchen attached to provide for serving luncheons.

On January 1 the following changes were made in the organization of the New Departure Mfg. Co.: Mr. Albert F. Rockwell, who was one of the founders of the company and who has taken an active part in the development of patents covering the product, has been relieved of certain managerial details, but continues as president of the company. Mr. DeWitt Page, who formerly held the positions of secretary, sales manager, purchasing agent and advertising manager, has been appointed general manager of the company. Mr. Charles T. Treadway, for some years past treasurer of the company, continues in that capacity, but also becomes chairman of the board of directors.

* * *

Tax factories and there will be fewer factories; tax trade and there will be less trade; tax machinery and there will be less machinery; but tax the value of land and of natural resources and there will not be any less land or natural resources.

A ROTARY CUTTING-OFF MACHINE

A machine which has accomplished some remarkable performances in cutting off bars and tubes has recently been built by Charles Taylor, Ltd., Birmingham, England. The cutter revolves around the bar or tube to be cut off, the work being held stationary in a vise. This avoids revolving long and heavy bar stock, and facilitates moving up and regripping the work. The cutters do not have to be stopped while the rod is being adjusted, thus reducing to a minimum the time lost in cutting off several pieces in succession.

The headstock consists of a substantial iron casting forming the main body of the machine and having massive split bearings in which runs a hollow spindle carrying the cutter-head. The cutter-head has two cutter-slides operated by a hand lever, which feeds the cutters in by means of chains passing through the spindle. These chains act in connection with an automatic balancing device to insure the cutters doing equal work. Fig. 1 shows how this is accomplished. In this illustration *A* is the cutter, *B* the slide carrying the cutter, *C* the chain which feeds the cutter in, and *D* a yoke which can oscillate on a cylindrical bearing in the piece *E*. Thus as one cutter is forced out, the yoke *D* moves on its bearing and draws the opposite cutter in until both cuts are exactly equal, and they remain so throughout the cut.

In most cutting-off machines it is highly important for both cutters to be set precisely alike to a gage, and this re-

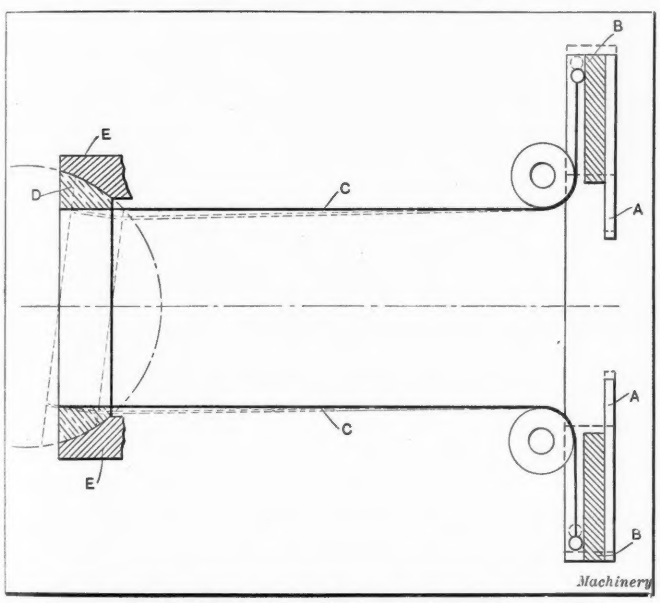


Fig. 1. Diagram illustrating Method of feeding Cutters to Work

quires a skilled operator. In the machine under consideration, this is not of great consequence because if the cutters *A* are carelessly placed, even to the extent of 1/16 inch or more, as indicated by the dotted lines on the end of the cutters, the effect when the machine is at work is that one cutter retreats and the other advances until both cutters are working precisely alike. When this balance is obtained the cutters remain in position until they are again taken out for resharpening. An unskilled operator can handle this balancing machine.

The feed chains are protected by a steel liner running the whole length of the spindle, the spindle being driven by a balanced cast-iron pulley of 16 inches diameter, for a three-inch belt. The cutters are of plain oblong section and are rigidly supported throughout their length. As shown in Fig. 2, they are so placed as to have ample clearance at all points and require grinding on one face only. This is accomplished by having their bearings so shaped that when the cutters are clamped in place they lie in such a position that only the extreme corner of one side of each is in contact with the work, except, of course, the front cutting edge. All other parts have a liberal clearance. Moreover, the cutters are offset from each other so that each takes a cut of less width than the groove which is being cut. This allows the chips to swell, as all chips do, without binding in the groove. Also when the cutters are dull they simply yield slightly from the face of the groove and so touch without the slightest binding effect. As the cutters are of plain rectangular section set to cut their own clearance, they cannot wear narrow on the front or cutting edge and bind in the groove, which is a frequent and continuous trouble with the ordinary formed cutting-off tool. In Fig. 3, for instance, showing the usual formed tool, the points *A* and *B* break off causing the tool to

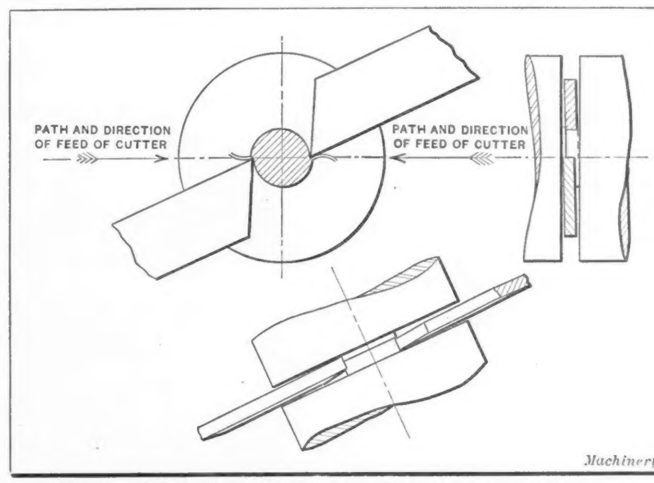


Fig. 2. Method of mounting Cutters that gives Ample Clearance

bind in the groove on the sides *C* and *D* with consequent breakage.

Other features of the machine include a lever stop with a spring plunger set to act in such a way that the spring is compressed as the stock is cut through, thus serving to retard the advance of the cutters at the last moment and to insure that the end of the stock is left clean. The stock is held in a self-centering vise in front of the cutters, and provision is made to support the stock which projects from the vise by means of an adjustable roller support, which facilitates the feeding of heavy stock through the vise. The cut off pieces are supported in the spindle on a long supporting member projecting into the spindle from the rear end. A two- or three-speed countershaft is provided enabling the speed to be changed, if necessary, while cutting is actually in progress.

The performance of this machine can be gaged as follows. The average time for cutting off solid bars is, for 3/4-inch round mild steel, 11 seconds; 1-inch round mild steel, 12 seconds; 1 1/2-inch square mild steel, 19 seconds; 2-inch round mild steel, 22 seconds; 3-inch round mild steel, 45 seconds; and 2-inch round cast steel, 65 seconds. In dealing with gas pipe, a 1 1/2-inch pipe can be cut off in 5 seconds. J.

* * *

DELICACY OF THE CHEMICAL BALANCE

Those not familiar with the methods of chemical analysis have very little conception of the delicacy of the chemical balance and extreme care that must be taken when ascertaining weights to a fraction of a milligram. The balance is supported by the knife-edges only when actual weighing is in progress. When the substance to be weighed is being placed on the pan, and during all manipulations of the weighing, the rule is to raise the beam off the knife-edges by means

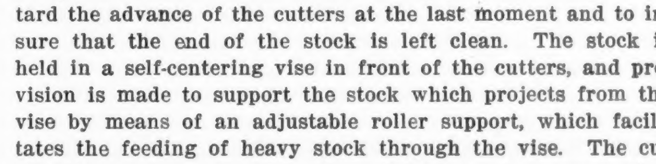
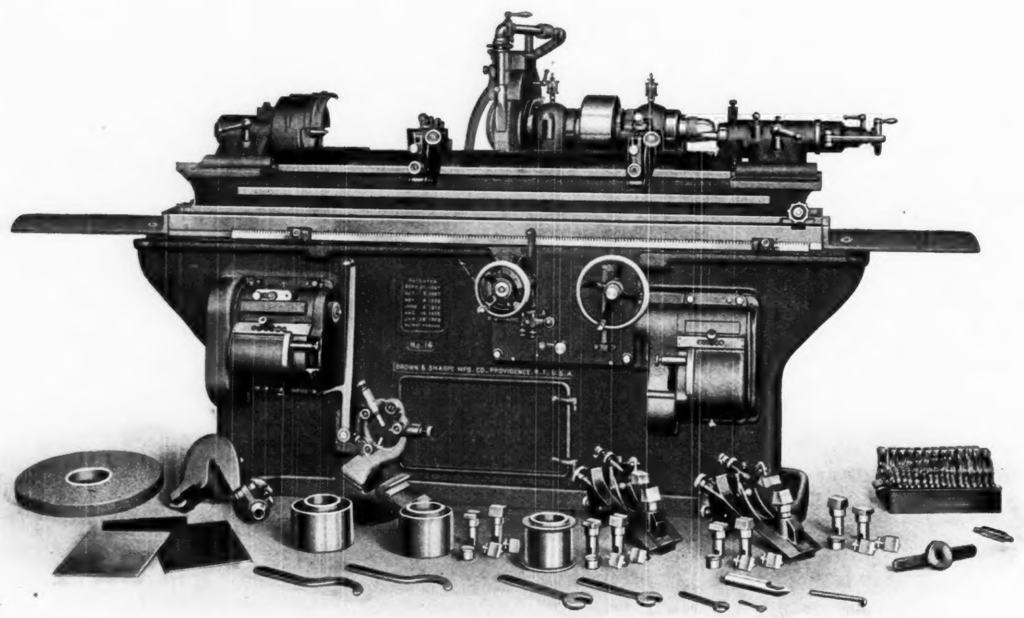


Fig. 3. Diagram illustrating Weakness of Formed Cutter

provided for the purpose. When not in use the beam is always supported by the auxiliary means provided. Extreme care must be taken to keep the knife-edges free from corrosion. A jar of unslacked lime or pure sulphuric acid is kept in the case to absorb atmospheric moisture. A warm object cannot be accurately weighed because the air currents set up a disturbing force sufficient to vitiate the accuracy of the work.

Grinding Time on This Work is



No. 14 PLAIN GRINDING MACHINE

You are probably familiar with some, if not all of the various types of machine parts represented in the case opposite.

It is the sort of work to be found in quantity in many manufacturing shops—work on which a small saving per piece means a worth while saving by the end of the day.

There are long, slender shafts requiring careful support, pieces of large diameter on which heavy cuts are taken, taper bearings on spindles, irregular shaped work as shown near top of case, etc.

We have developed the type of Plain Grinding Machine with quick change work speeds and table traverse, shown above, as particularly fitted for rapid production on such work.

BROWN & SHARPE

PROVIDENCE, R. I.,

OFFICES: 20 Vesey St., New York, N. Y.; 654 The Bourse, Philadelphia, Pa.; 626-630 Washington Blvd., Chicago, Ill.; 305 Chamber of Commerce Bldg., Rochester, N. Y.; Room 429 University Block, Syracuse, N. Y. **REPRESENTATIVES:** Baird Machinery Co., Pittsburgh, Pa., Erie, Pa.; Carey Machinery & Supply Co., Baltimore, Md.; E. A. Kinsey Co., Cincinnati, Ohio, Indianapolis, Ind.; Pacific Tool & Supply Co., San Francisco, Cal.; Strong, Carlisle & Hammond Co., Cleveland, O., Detroit, Mich.; Colcord-Wright Machinery & Supply Co., St. Louis, Mo.; Perine Machinery Co., Seattle, Wash.; Portland Machinery Co., Portland, Ore. **CANADIAN:** The Canadian Fairbanks-Morse Co., Ltd., Montreal, Toronto, Winnipeg, Calgary, Vancouver, St. Johns, Saskatoon. **FOREIGN:** Buck & Hickman, Ltd., London, Birmingham, Manchester, Sheffield, Glasgow. F. G. Kretschmer & Co., Frankfurt, a/M., Germany. V. Lowener, Copenhagen, Denmark; Stockholm, Sweden; Christiania, Norway. Schuchardt & Schutte, St. Petersburg, Russia. Fenwick Freres & Co., Paris, France; Liege, Belgium; Turin, Italy; Zurich, Switzerland; Barcelona, Spain. The F. W. Horne Co., Tokio, Japan. L. A. Vail, Melbourne, Australia. F. L. Strong, Manila, P. I.

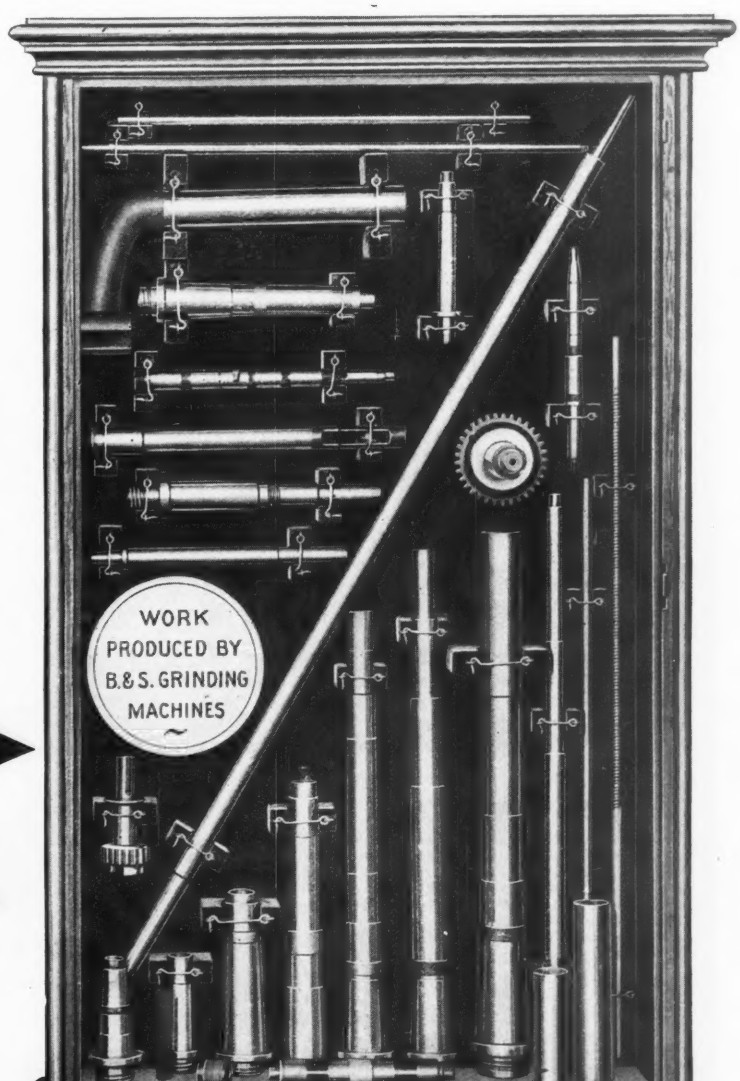
Reduced By Our New Machine

When the foreman of the grinding department distributes work similar to that shown in the case, he often gives advice as to the approximate rate of work speed and table traverse to be used. But it lies with the operators to try these suggestions and change about, if necessary, until the combination best suited to each particular job is obtained. A slight change in speed or traverse of the work means a considerable difference in the finish, as well as the rate of production.

We have made it handy for the operators to change speeds and feeds, consequently they will adjust the combinations to get the fastest production consistent with the required finish.

The speed and feed changing mechanisms are both self-contained in the machine—no complicated overhead works. Changes are instantly made by adjustment of index slide and levers. The two change gear boxes are located right beside the operator. He can change from a roughing to a finishing feed by the movement of one lever. Ask us about some of the other rapid production features.

The Combination Plain and Universal Back Rests furnish a support which allows rapid grinding on all varieties of work. For long, slender shafts, either with or without keyway, they are used as full universal rests, following up the work as it is reduced in diameter. By substituting a solid, adjustable shoe beneath the work, they are made semi-universal, for use on heavy work. For the very heavy grinding, both spring actuated shoes are changed to solid adjustable ones, making them Plain Back Rests.



MFG. CO.

U. S. A.

Read Page 67

PERSONALS

Henry Pollard recently took charge as superintendent of manufacturing of the Chicago plant of the Benjamin Electric Co.

E. E. Hendee, secretary of Joseph T. Ryerson & Son, returned to Chicago in the middle of February after a two weeks' trip to the South and Southwest.

Edward Blake, who for several years was sales manager of Wells Bros. Co., Greenfield, Mass., and for the past three years general manager of the J. T. Slocomb Co., Providence, R. I., has resigned his position with the company. Mr. Blake has made no definite plans for the future.

A. B. Hall, for the past fourteen years connected with the machinists' supply department of the Whitman & Barnes Mfg. Co., was elected a director and second vice-president of the company at the annual stockholders' meeting held January 28 at the general offices, Akron, Ohio.

William Miller has discontinued his connection as secretary of the firm of Henry Disston & Sons, of Philadelphia, after a service of thirty-seven years, and on February 1 associated himself with the Simonds Mfg. Co., Fitchburg, Mass., and Chicago, Ill. Mr. Miller is one of the most experienced and efficient men in the saw, knife and file-making business, and is well and favorably known in the hardware trade.

O. P. Wilson, of the purchasing department of the Westinghouse Electric & Mfg. Co., has resigned to take the position of assistant general manager of the Norma Company of America, 20-24 Vesey St., New York City, manufacturer and importer of high-grade ball bearings, precision instruments, etc. Mr. Wilson has been connected with the Westinghouse Electric & Mfg. Co. for fourteen years, and for the last ten years has been one of its head buyers.

C. S. Dundore, after thirteen years' service as treasurer and general manager of the American Die & Tool Co., Reading, Pa., has started in business for himself under the name of the Dundore Mfg. Co., 221 S. 9th St., Reading, Pa. The company will serve the tool and contract work trade, being equipped to make tools and dies and also to do contract work in the manufacture of machines and metal specialties. Special attention will be given to the manufacture of boiler and bridge builders' tools.

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OBITUARIES

Neil W. Snow, president and general manager of the Detroit Twist Drill Co., Detroit, Mich., died January 22, aged thirty-four years.

William Colthar, proprietor and general manager of the Victor Vise Co., Springfield, Ohio, and a manufacturing machinist, died January 19, aged forty-nine years. Mr. Colthar acquired his mechanical training with the Mechanics Institute at Cincinnati, and after considerable experience in the machine tool industry in the latter city he undertook work on his own account, being the original designer of the Timken roller bearing. He did considerable work on the Paige typesetting machine, and designed many original machines. He is survived by his widow and one son.

Caleb Colvin, founder of the Caleb Colvin iron foundry, which later became the L. W. Pond Machine & Foundry Co., Worcester, Mass., died of bronchitis at his home in Worcester, February 17, aged eighty-five years. He was born in Cranston, R. I., and at the age of eighteen was apprenticed to the iron molder's trade. He went to Worcester in 1865 and established the iron foundry business, which was afterward made a partnership known as C. & J. A. Colvin. The business was incorporated in 1887 under the name of the L. W. Pond Machine & Foundry Co. Mr. Colvin retired from active business in 1905.

Erwin Starr Sperry, editor and publisher of the *Brass World and Platers' Guide*, died at Bridgeport, Saturday, January 31. Mr. Sperry was born in Ansonia, Conn., in 1866, and graduated from the Sheffield Scientific School of Yale University in 1887, where he afterward held a position as assistant instructor in chemistry under Prof. H. L. Wells. In 1891, he went to Bridgeport as chemist for the Aluminum Brass & Bronze Co., and afterward became superintendent of the Waldo Foundry. Ten years ago, he started publishing the *Brass World and Platers' Guide*. He was a member of leading scientific societies of this country and Europe. Mr. Sperry is survived by his widow.

David B. Hyde, one of the Hyde brothers who have been closely connected with the emery wheel manufacturing business since its inception, died February 14 at Riverside, Cal., of pneumonia, aged fifty-five years. He was born at Wilmington, Vt., in 1859 and at the age of thirteen was thrown upon his own resources. He and one of his brothers (mere boys at the time) showed great enterprise by placing on the market a patent inkstand, and thousands of these stands were sold in America and Europe. Mr. Hyde was one of the pioneers in the manufacture of emery wheels and grinding machinery. In 1880 he, with his four brothers and D. T. Homan, started the Springfield Glue & Emery Wheel Co. at Springfield, Mass., being the first to make emery wheels for water tool grinding in this country. In 1893 he and his brother, O. H. Hyde, and E. C. Gwynn, organized and started the Safety Emery Wheel Co. of Springfield, Ohio, and he introduced the safety collars used on emery wheels. Mr. Hyde also originated the "Champion" tool-holder now manufactured by the Western Tool & Mfg. Co. of Springfield, Ohio. He was later connected with the Pittsburg Emery Wheel Co., but on account of poor health sold out his business interests and went to California, where he was engaged in dry farming on an extensive scale at the time of his death. Mr. Hyde leaves a widow, two daughters and one son; also three brothers, E. R. Hyde of the Bridgeport Safety Emery Wheel Co. and O. H. and C. L. Hyde of the Safety Emery Wheel Co. of Springfield, Ohio.

Henry Brinton, president of the H. Brinton Co., Philadelphia, Pa., died at his home at Bala, January 30, after a brief illness with heart trouble, aged sixty-six years. He was born at Christiana, Lancaster Co., Pa., of a line of Quaker ancestry descended from William Brinton who came to America in 1684. He received a thorough training as a machinist in the shops of I. Broomell & Sons, now the Christiana Machine Co., and later spent about two and one-half years with Bement-Miles & Co., Philadelphia. From there he went to the Colt's Armory in Hartford, Conn., where he worked for a time on the Baxter engine. His next work was on the Branson knitting machine at Bellefonte, Pa. When the Branson Co. failed, Mr. Brinton went to the Lancaster Watch Co. at Lancaster, Pa., where he obtained experience in fine manufacturing methods which had a pronounced effect on his own designing later. When the Branson Co. was reorganized, Mr. Brinton went back to Bellefonte, only to meet another financial crisis. After a further experience at the Lancaster Watch Co., he joined Branson a third time in Philadelphia and was for some years the superintendent of the knitting machinery business under the Branson name. In 1888 Mr. Brinton started manufacturing knitting machines in the partnership of Brinton, Denney & Co., and five years later, Mr. Denney sold his interests to J. E. Longergan and the firm name became H. Brinton & Co. The business was incorporated in 1906 under the name H. Brinton Co. Mr. Brinton probably did more to improve and commercialize automatic cylinder knitting machines of the latch needle type than anyone else. He took out patents on the knitting machine sinker mechanism with the sinker driven from below the stitch level. On ribbers, the Brinton design was revolutionary. The ribber pattern wheel mechanism built by Mr. Brinton twenty years ago, has not been surpassed or changed to this date. Splicing devices for inserting an extra thread is also of Brinton design and a ring in the dial cap for operating multiple-feed rib machines was another Brinton invention. In general, it may be said that the prime feature of Mr. Brinton's work was the beautiful simplicity of his mechanism for producing complex results. When he finally adopted a design, that design was usually as near to the perfection of mechanical simplicity as it was possible for it to be made. He was married in 1874 to Rachael Cawley, who with a daughter and two sons survive him.

COMING EVENTS

April 4-11.—First National Efficiency Exposition and Conference, Grand Central Palace, New York City. Walter H. Tallis, director, Efficiency Society, Inc., 41 Park Row, New York City.

April 29-30.—Annual meeting of the National Association of Cotton Manufacturers, Boston, Mass., in the Paul Revere Hall of the Mechanics Bldg. C. J. H. Woodbury, secretary, 45 Milk St., Boston, Mass.

May 1-October 31.—Anglo-American Exposition, London, England, to celebrate the centenary of peace between the United States and Great Britain. American executive offices: Woolworth Bldg., New York City. Charles J. Kiralfy and Albert E. Kiralfy, commissioners general.

June 15-17.—Annual convention of the American Supply & Machinery Manufacturers' Association at White Sulphur Springs, West Virginia; New Green Brier Hotel, headquarters. General offices of the association, Woolworth Bldg., New York City.

September 20-25 (1915).—International Engineering Congress, San Francisco, Cal., in connection with the Panama-Pacific International Exposition. W. F. Durand, chairman, Foxcroft Bldg., San Francisco, Cal.

NEW BOOKS AND PAMPHLETS

Copper Wire Tables. 69 pages, 7 by 10 inches. Published by the Department of Commerce, Washington, D. C., as Circular 31 of the Bureau of Standards.

Standard Specifications for Incandescent Electric Lamps. 20 pages, 7 by 10 inches. Published by the Department of Commerce, Washington, D. C., as Circular 13 of the Bureau of Standards.

United States Coast and Geodetic Survey. Annual Report of the Superintendent to the Secretary of Commerce. 100 pages, 6 by 9 inches. Published by the Department of Commerce, Washington, D. C.

High-frequency Ammeters. By J. H. Dellinger. 70 pages, 7 by 10 inches. Published by the Department of Commerce, Washington, D. C., as Reprint No. 206 from the Bulletin of the Bureau of Standards, Vol. 10.

Electrolytic Corrosion of Iron in Soils. By Burton McCollum and K. H. Logan. 69 pages, 7 by 10 inches. Published by the Department of Commerce, Washington, D. C., as No. 25 of the Technologic Papers of the Bureau of Standards.

Latent Heat of Fusion of Ice. By H. C. Dickinson, D. R. Harper and N. S. Osborne. 32 pages, 7 by 10 inches. Published by the Department of Commerce, Washington, D. C., as No. 209 of the Scientific Papers of the Bureau of Standards.

Coal Washing in Illinois. By F. C. Lincoln. 108 pages, 6 by 9 inches. Published by the University of Illinois, Urbana, Ill., as Bulletin No. 69. Price 50 cents.

A great amount of data gathered from experiments made in the mining laboratory of the university, personal visits to the various coal washeries in the state and a thorough search of the scattered literature on the subject has been compiled and tabulated in this bulletin. The bulletin also contains an article by S. W. Parr on the moisture in washed coals and is a summary of tests made at the Experiment Station on carload lots of various washed coals.

Elementary Manual of the Steam Engine. By Ernest V. Lallier. 266 pages, 5 by 7¾ inches. 102 illustrations. Published by D. Van Nostrand Co., New York City. Price $2.

This work, as indicated by the title, is intended for the instruction of students and young men studying the elements of engineering. It treats of reciprocating steam engines, governors, engine calculations, the indicator, heat, boilers, pumps, Corliss engines, pipes and fittings, rotary engines, internal combustion engines and lubrication. Wherever possible, principles are illustrated with arithmetical examples, and questions are given at the end of chapters which may be used for self-examination to the student studying at home. The work is one that should be appreciated by the class generally for whom it is intended.

Factory Organization and Administration. By Hugo Diemer. 378 pages, 6 by 9 inches. 172 illustrations. Published by the McGraw-Hill Book Co., Inc., New York City. Price $3.

The first edition of this work was published in 1910. In rewriting the book, the author has rearranged the material in what appears to be more logical order. The subjects treated comprise: Principles, Field and Methods of Industrial Management; Industrial Finance; Organization and Control; Typical Factory Organizations; Factory Accounts; Departmental Reports; Factory Location; The Planning of Factory Buildings and the Influence of Design on Their Productive Capacity; Employment of Labor and Labor Problems; General Office; Order Department; Bills of Material; Drafting Department; Pattern Department; Purchasing Department; Stores and Stock Department; The Production or Planning Department; Foundry Systems; The Machine Shop and Tool Department; Shipping and Receiving Departments; Time Taking; Cost Department; Aids in Taking Inventory; Inspection Methods in Modern Machine Shops; Rate Fixing and Time Studies; Wage Systems; Principles Underlying Good Management; and A Bibliography of Works Management.

Welding—Theory, Practice, Apparatus and Tests. By Richard N. Hart. 210 pages, 6 by 9 inches. 127 illustrations. Published by the McGraw-Hill Book Co., Inc., New York City. Price $2.50

The first edition of this work was published in 1910. The rapid progress of the last three years in apparatus and methods of welding has made a revision necessary, especially with regard to arc welding and oxy-acetylene welding. The work deals with the characteristics of iron, platinum, gold, silver, aluminum, copper, and nickel. The electric welding processes are described, including the La Grange-Hobo process, the Zerener electric blowpipe, the Bernardos arc-welding process, the Slavianoff arc-welding process, combination Slavianoff and Bernardos processes, and the Thomson process. Part III is devoted to hot-flame welding, treating of the oxy-acetylene and oxy-hydrogen processes. The torches and apparatus in general are illustrated and described. A chapter is devoted to the thermit process, illustrating its apparatus and application to rail welding, marine repairs and other characteristic jobs for which it is especially suited. A chapter is given up to soldering and miscellaneous processes including the Lafitte welding plate, the "ferrofix" brazing process, brazing and soldering, the Blaugas process, etc. The work is one to be recommended to all desiring to become acquainted with the characteristics of the various welding processes developed within the past few years. While the treatment of the subjects is by no means exhaustive, it leaves comparatively little to be desired by those who wish to gain a general knowledge of the modern welding art.

NEW CATALOGUES AND CIRCULARS

Massachusetts Institute of Technology, Boston, Mass. Catalogue for 1913-1914.

Stevens Institute of Technology, Hoboken, N. J. Annual catalogue 1914-1915.

Technical Supply Co., Scranton, Pa. Booklet describing the Levi automatic blueprint finisher.

Polytechnic Institute, Brooklyn, N. Y. Annual catalogue of the College of Engineering, 1914-1915.

Alston Saw & Steel Co., Folcroft, Pa. Circular of the Alston process improved hacksaw blades made for all purposes.

Laidlaw-Dunn-Gordon Co., Cincinnati, Ohio. Bulletin 22 on air compressor efficiency and the factors controlling it, with special reference to the Cincinnati air gear.

Brown Hoisting Machinery Co., Cleveland, Ohio. Pamphlet C on "Brownhoist" safety crabs and winches. Provision is made so that the handles cannot fly back and the load cannot drop.

Hannifin Mfg. Co., Chicago, Ill. Circular illustrating and describing Hannifin air-operated chucks, including master hinge air-operated collet chucks, and two- and three-jaw universal chucks.

H. A. Lowe, 1374 E. 88th St., Cleveland, Ohio. Circular of the Lowe universal test indicator weighing 1¼ ounce and magnifying 100 times. The dial is graduated to thousandths inch.

Northern Engineering Works, Detroit, Mich. Crane catalogue 26 illustrating electric traveling cranes, hand power traveling cranes, electric and pneumatic hoists, overhead track systems, bucket handling cranes and railway cranes.

Railway Roller Bearing Co., Syracuse, N. Y. Catalogue of the roller journal boxes, electric motor bearings, etc. The construction of the bearing is shown and examples of applications to electric locomotives, street cars, motors, etc., are included.

Simplex Wire & Cable Co., 201 Devonshire St., Boston, Mass. The "Simplex" manual containing, in addition to information regarding "Simplex" products, tables and data for the ready reference of electrical engineers, contractors, wiremen, etc.

Greenfield Machine Co., Greenfield, Mass. Catalogue No. 5 of the "Greenfield" universal tool and cutter grinder and its attachments. The uses of the attachments are illustrated and other matter is contained of value to users of tool grinding machines.

Mesta Machine Co., Pittsburg, Pa. Bulletin M of the Mesta improved pickling machines for pickling metal objects of any shape. The Mesta machine brings mechanical action into play to such an extent that the material is pickled with about one-half of the acid and labor required in hand pickling.

Webster & Perks Tool Co., Springfield, Ohio. Circulars of Nos. ¾ floor or bench type grinder, 1¼ "T" self-oiling grinding and polishing machine, 1⅜, 1½ and 1⅝ direct-current electrically driven floor type polishing and buffing lathes, and 1¾ alternating-current direct-connected electrically driven type grinder.

Gould & Eberhardt, Newark, N. J. Catalogue of high-duty shapers and attachments, details of construction and general specifications. The line comprises 14-inch, 16-inch, 20-inch, 24-inch and 28-inch sizes. Shapers with direct-current and alternating-current motor drives are shown, and accessories for same.

Winfield Electric Welding Machine Co., Warren, Ohio. Catalogue of Winfield electric welding machines, comprising spot and butt welding machines, which are illustrated and described. The catalogue contains information on electric welding useful to manufacturers employing the electric welding process.

Royersford Foundry & Machine Co., 54 N. 5th St., Philadelphia, Pa. Catalogue "Saving the Friction Loss—'Sells' Roller Bearings," containing a description of the construction, operation and application of "Sells" roller bearings. The catalogue is illustrated with halftone and sectional views, the latter showing the design in detail.

Wood Turret Machine Co., Brazil, Ind. Catalogue of the "tilted turret" lathe, a feature of which is that the turret is so made and placed that the stock can pass clear through, thus giving unlimited capacity in length of work. The machine is made in six sizes, ranging from 1 inch to 4½ inches automatic chuck capacity.

National Lead Co., 111 Broadway, New York City. Pamphlet on electro-plating zinc alloy die-castings, giving practical methods for cleaning, polishing and electro-plating the various alloys used in die-casting. The information contained will be particularly acceptable to electro-platers who have experienced difficulty in doing this class of work.

Pawling & Harnischfeger Co., Milwaukee, Wis. Pamphlet entitled "Difficult Drilling and Boring Made Easy," illustrating work done by the Pawling & Harnischfeger drill presses in machine shops. Manufacturers having to drill large structural work, castings, etc., should find the illustrations and descriptions of these drilling machines of interest.

National Tube Co., Pittsburg, Pa. Bulletin No. 18-A entitled "National Reamed and Drifted Pipe." This bulletin contains a complete description with illustrations of this product, together with a short introduction explaining the processes of well drilling and information relative to the various accessories necessary for the drilling and pumping of wells.

Cutler-Hammer Mfg. Co., Milwaukee, Wis. Pamphlet on door safety switches for use with electric elevators. A great majority of accidents on all kinds of elevators are the result of setting the elevator machine in motion before the doors are closed and locked and the passengers clear. With the ever-increasing number of passengers the safety feature is of the utmost importance.

National Tube Co., Frick Bldg., Pittsburg, Pa. Bulletin 19, listing the products of the company. An idea of the extent of the products manufactured can be gained from the fact that the bulletin has eight pages, two columns to the page, filled with small type. Of "Kewanee" products, 852 are listed; of "National" pipe products, sixty-four, and of Shelby seamless steel tubing products, thirty-seven.

Niles-Bement-Pond Co., 111 Broadway, New York City. Catalogue of Niles electric traveling cranes, showing the equipment of Niles standard traveling cranes and details of construction. Grab bucket trolleys, electric wall cranes, gantry cranes, traveling hoists and hand power frames are also shown. The catalogue is beautifully illustrated with numerous views of works in which Niles bridge traveling cranes are installed.

Curtis Pneumatic Machinery Co., 1568 Kienlen Ave., St. Louis, Mo. Catalogue of air compressors, air hoists, air cranes, pneumatic and hydro-pneumatic elevators, trolleys, trolley systems and sand blasts. The company has specialized in pneumatic hoisting appliances for over twenty years and has developed the simple air cylinder into a straight-line motor with wonderful speed control and dependability, capable of the widest application to hoisting problems.

Gardner Governor Co., Quincy, Ill. Circular "The Gardner One Tool Plant," consisting of a vertical self-oiling air cooled compressor, 2 by 2½ inches, driven by ½ H. P. electric motor, mounted on a four-wheel truck. A tool-box is provided and a rack for hose. The machinery is enclosed in a removable cover and the outfit forms a compact, portable compressed air plant. While primarily designed for stoneworkers, monumental work, etc., it should serve in many other situations where compressed air is required.

Yale & Towne Mfg. Co., 9 East 40th St., New York City. Pamphlet entitled, "History of the Trade-mark 'Yale,'" which is a record of cases in which the company's rights have been sustained and confirmed in litigation involving a legal right in trademarks, catalogue numbers, distinctive designs and other indications of the origin of patterns. The pamphlet is one that should be read with interest by all concerned with trademarks, and it should also be found interesting by readers in general because of the matter contained on the origin of locks.

Titanium Alloy Mfg. Co., Niagara Falls, N. Y. Rail Reports Bulletin No. 4 on open hearth steel. This bulletin contains a summary of chemical and physical results as reported in bulletins 1, 2, 3 and 4 on standard open hearth A rails and titanium open hearth treated A rails. It was found that the average hardness of standard rails is twenty-four per cent greater in the webs than in the heads and flanges while in the titanium treated rails, the average difference is only five per cent. The impact resistance of titanium rails averaged thirty-five per cent higher than of standard rails in the heads, four per cent in the webs and fifty per cent in the flanges. Titanium rails in the White-Souther endurance test averaged fifty per cent greater endurance than standard rails. The value of these reports on extensive rail tests is evident to all metallurgists and users of rails.

H. W. Caldwell & Son Co., Western Ave., 17th to 18th Sts., Chicago, Ill. Catalogue No. 38 on elevating, conveying, power transmitting and general machinery, containing 797 pages, 6 by 9 inches. The company makes helicoid conveyors, machine molded gears, friction clutches, ice handling machinery, specialties for flour mills, grain elevators, cotton seed oil mills, alfalfa plants, starch works, linseed oil mills, breweries, distilleries, malt houses, sugar refineries, glucose works, cement works, phosphate works, lime works, gypsum works, chemical works, tanneries, etc. The catalogue is one of the most complete in the line of power transmission elevating and conveying machinery. It is copiously illustrated, printed on high-grade paper and completely indexed. It should be in the hands of every engineer and manufacturer having use for the line of machinery and parts listed.

R. K. LeBlond Machine Tool Co., Cincinnati, Ohio. Catalogue of LeBlond engine lathes which are furnished in various styles, with such combinations of equipment as may be desired, as follows: regular stud lathe, 14 to 20 inches swing; regular standard engine lathe, 12 to 24 inches swing; regular quick-change engine lathe, 12 to 24 inches swing; heavy-duty manufacturers' automobile lathe, 17 to 21 inches swing; seventeen-inch rapid production lathe; heavy-duty standard engine lathe, 25 to 33 inches swing; heavy-duty quick change engine lathe, 17 to 33 inches swing; heavy-duty quick change engine lathe, 25 to 33 inches swing; heavy-duty quick change sliding bed gap lathe, 19-38, 25-50 and 30-60 inches swing; regular plain chucking lathe, 12 to 24 inches swing; heavy-duty quick change combination turret lathe, 17 to 27 inches swing; heavy-duty manufacturers' complete turret lathe, 24 to 27 inches swing—with triple gear single pulley drive, 31 inches swing; heavy-duty quick change universal turret lathe, 17 to 33 inches swing; motor-driven engine lathes of any size or type. A line of accessories is shown comprising turrets, turret toolposts, compound rests, relieving attachment, multiple automatic length stop, belt shifter, draw-in attachment and collets, translating gears for metric threads, etc.

TRADE NOTES

Warren Forge & Tool Co., Warren, Ohio, has elected James Robertson, president; M. J. Konold, vice-president and secretary; George E. Warner, treasurer; and George F. Konold, general manager.

Walcott & Wood Machine Tool Co., Jackson, Mich., has bought the patent rights, patterns and good-will of the Melling-Northrup die-sinking machine manufactured by the Melling-Northrup Co., Jackson, Mich.

Gem City Machine Co., 429 East First St., Dayton, Ohio, was destroyed by fire during the last week in January. Three days later, however, the firm was operating in other quarters, delivering orders and contracting for new work.

Buffalo Forge Co., Buffalo, N. Y., has opened offices at 176 Federal St., Boston, Mass., for its fan, ventilating and pump department. The company is represented in the New England territory by B. R. Andrews, formerly with the B. F. Sturtevant Co.

J. T. Slocomb Co., Providence, R. I., maker of machinists' micrometers, centering drills, etc., has been sold to J. H. Drury, sales manager of the Union Twist Drill Co., Athol, Mass. J. T. Slocomb, who organized the business, will remain for a short time.

Eveland Engineering & Mfg. Co., 2324-2328 Market St., Philadelphia, Pa., has been authorized by its

board of directors to increase the capital stock from $100,000 to $2,000,000 for the purpose of extending its business of manufacturing machine tools, riveters, etc.

Nutter & Barnes Co., manufacturer of metal saw cutting-off machines, automatic saw sharpeners, abrasive wheel cutting-off machines, has moved from Boston, Mass., to Hinsdale, N. H., where a larger and better factory than the one occupied in Boston has been equipped.

Vulcan Engineering Sales Co., Mumford Molding Machine Co., Hanna Engineering Works, and the **Q. M. S. Co.** have moved their general sales offices to their factory at 2059 Elston Ave., Chicago, Ill. At this address a large stock of foundry, riveting and metal working machinery is carried.

Standard Machinery Co., 7 Beverly St., Providence, R. I., builder of presses, drop hammers, rolling mills, wire drawing machinery, rotary swaging machines, dies, etc., has moved from Providence, R. I., to its new plant at Auburn, R. I., located five miles from the former plant in Providence.

Racine Electric Co., Racine, Wis., manufacturer of small electric motor specialties such as bench, internal and center grinders, fans, etc., has opened general offices in the Fischer Bldg., Chicago, Ill., to which office all correspondence should be addressed. J. A. Brown, formerly of the sales department, is general manager and W. C. Perkins, assistant manager. The concern was started in a small way a few years ago and now has a rapidly growing business.

Century Electric Co., St. Louis, Mo., manufacturer of alternating-current motors and fans, is now located in its new building, 19th and Pine Sts. The new building is eight stories and basement, reinforced concrete, and is connected with the company's old plant by three bridges and tunnels. The general offices are in the new building. The addition doubles the company's floor space. This concern, which was started only four years ago, has shown a remarkable growth.

International Money Machine Co. was recently incorporated in the state of Indiana. The company is not a new enterprise, but is incorporated to take over and operate the business of the United States Cashier Co., of Portland, Oregon. The growth of the business of the latter concern has necessitated the removal of the plant to a more central location. Frank Menefee, formerly of Portland, Oregon, is president of the new organization which is incorporated with a capital of $2,000,000.

Brown-Wales Co., Boston, Mass., is now the exclusive distributor in New England of Toledo high-speed and carbon tool steels, the firm of Harrington, Robinson & Co., who has previously handled this line, having discontinued the sale of high-grade tool steels and simultaneously the agency for the Toledo steels. Mr. W. T. James, who has been the chief Toledo traveling representative for Messrs. Harrington, Robinson & Co., will continue in a like capacity, traveling from the New York office of the Newman-Andrew Co., 107 West St., New York, chief representatives of the Toledo Steel Works, Sheffield, England.

Hermann Boker & Co., New York City, are now in a position to make stock shipments from their new Cleveland warehouse, 703 Frankfort Ave., N. W., Cleveland, Ohio. They have there in stock the same regular lines of high-speed and carbon tool steels, nickel bars, sheets and rods, music wire, steel balls, etc., as carried in their other warehouses in this country. The Cleveland office is in charge of Wilmot H. Kissam, who for many years was connected with the New York office. The territory covered by the Cleveland office and visited by its representatives will include Ohio and parts of West Virginia and Kentucky.

Scoville Foremen's Association, Inc., Waterbury, Conn., held its third annual dinner January 28, at the Hotel Elton, at which time the principal address was made by Prof. Herman Schneider, dean of the College of Engineering of the University of Cincinnati, Ohio, who spoke interestingly on the different characteristics of engineering students and of the work done in Cincinnati to educate engineers simultaneously along practical and theoretical lines. Besides Prof. Schneider there were several other speakers, including Mayor Martin Scully and Rev. E. J. Brennan. Mr. E. S. Sanderson acted as toastmaster. One of the features of the banquet was the attractive menu card enclosed in a cover of sheet brass rolled in the Scoville Mfg. Co.'s factory.

Lufkin Rule Co., Saginaw, Mich., announces that it has improved its general line of steel tapes by providing a steel case liner for leather cases and a push-button opener for the winding handles. These features have usually been found heretofore only on the highest priced steel measuring tapes. The company's "Challenge" and "Challenge Junior" steel tapes now have leather cases steel lined throughout. This gives the case stability and also makes it possible to have it narrower than before by a full 1/8 inch. The "Rival" and "Rival Junior" steel tapes have nickel plated steel cases as before but the edge or case band is knurled to afford a firm hand-hold. The cases of these tapes also are now equipped with a positive action winding handle opener, the same as the "Challenge" and "Challenge Junior" mentioned before.